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Jig Design for the Staircase Auger

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Abstract

This is a jig design to operate the staircase auger. The jig is designed to test the effectiveness of the staircase auger chip removal theory while under Earth's gravitational pull and atmosphere. An outside company will provide the auger to be tested.

The jig assembly will generate a reciprocating motion and a rotation to the auger. The stroke motion will operate within a range of 1 inch to 6 inches at a frequency of 0-100 cycles per minute. Frequency is adjusted through changing the diameters of the pulleys used. The stroke power is supplied from a 1725 rpm, 1/3 horsepower AC motor. The rotational motion is supplied by a separate and smaller motor.

For mechanical simplicity a scotch yoke linkage design is employed with a rotating disk and roller to supply the reciprocating motion. The shaft from the yoke is coupled to a second shaft which will allow rotational motion. The auger is attached to this shaft. The whole assembly can be operated from an ordinary electrical outlet.

The effectiveness of the chip removal theory can be determined by measuring the amount of dirt brought out by the auger in a testing environment.

Problem Statement

The objective of this project is to design a jig for the staircase auger to be used for tests to simulate lunar drilling. The auger used in the tests will be supplied. The jig must provide a linear stroke as well as rotational motion to prove the feasibility of the staircase auger theory. The frequency and length of auger strokes must be easily controlled along with the rotational speed of the auger.

Constraints

- The jig should be functional for many tests.
- The jig should be easily transportable and easily assembled.
- Stroke length range should be 1 to 6 inches. Stroke frequency should be 0-100 cycles/minute.
- Costs must be kept low.
- The jig should be safe to use.

Description

Introduction

The staircase auger has been proposed as a tool to be used on the moon to remove shavings during drilling into the moons surface. The design of the auger as proposed by previous ME 4182 groups has been put aside for a readily available model with different physical characteristics. In either case, the design allows the tip of the auger (referred to as the bit) to bore into the moon's surface, and for the cuttings to be lifted away by a reciprocating longitudinal motion of the staircase auger. One helical ramp and staircase steps make up the design of the device. It is believed that the combination of the rotary and vibratory motions will shake the cuttings up successive steps on the auger until they eventually reach the surface and are spilled out. It is the objective of this design group to demonstrate just how effective this idea is.

It is the design group's goal to demonstrate whether or not this theory will prove to be a satisfactory approach to remove cuttings from a bored hole. The auger and supporting constraints have been developed and established for us by previous ME 4182 design groups. However, we will be demonstrating the device's performance using an auger design from an outside source. The helix angle of this auger is 25°, and utilizes a single, helical ramp 30" long.

The first step in designing the testing system was to investigate different ways of producing the motions needed. The rotary motion needed was a secondary consideration, so it did not present a problem, however, the reciprocating motion combined with the rotary motion did give us some difficulty. The introduction of reciprocating motion combined on the same shaft as rotary provided an engineering problem. A way of coupling these two motions needed to be considered. Also the design constraints specified a square-wave type waveform to be used. This on-off type motion is extremely difficult to reproduce, so a steeply sloped sine-wave was used as an input waveform. We considered variations of three different machines to perform the necessary movement. The design given final approval was one utilizing a scotch yoke.

Mechanical System

Overview

The jig construction is designed for versatility, low cost, and effectiveness in demonstrating the staircase auger's performance. Since the design and the construction of the staircase auger have been supplied by outside sources, we will only be responsible for constructing the mechanism which will reproduce the type of motions prescribed by the designers of the auger.

Our initial considerations for building the jig are few, but important to the success of the design. Briefly, these include:

- •ability to reproduce motions faithfully.
- •availability of parts to build.
- •low cost.
- •simplicity of design.
- •adjustable features.
- •120V operation.

Our preliminary design ideas included a drill press type mechanism and a hydraulic mechanism. The drill press design is shown in Figure 1. The reciprocating motion was generated by the power screws. The hydraulic design (Figure 2) utilized a double acting hydraulic cylinder which provided both rotary and reciprocating motion. We dropped these ideas in favor of the scotch yoke mechanism.

Scheme of Operation

The mechanical system designed to drive the auger is based on a scotch yoke linkage. The auger is attached to a vertical shaft yoke, and a rotating disk and roller bearing produce a linear oscillating motion in the yoke (Figure 3). The disk is driven by a small motor using belts and pulleys. Rotational motion is provided from a separate motor. The advantage of this design is its mechanical simplicity and low production cost.

The principle idea behind the scotch yoke design is converting rotational motion to linear motion. A horizontal supported shaft and the disk are welded together. The shaft is supported by two pillow blocks. These bearings have set screws which lock down and hold the shaft in place so that only rotation is possible. The pillow blocks are bolted into a wooden member. The wooden member is part of the structure which supports the entire system. (see figure 4) The shaft supports most of the weight of the system on one end, however, bending stresses are negligible due to the strength of the shaft (Appendix 2).

The steel disk is supported by the horizontal shaft. Mounted on the disk is a sphere roller bearing. A single bolt is used to secure the bearing to the disk so that it can be easily removed. The diameter of the the bolt is 9/16". Here also, the stresses on the bolt are a small fraction of the bolt strength. The bearing supports the steel yoke.(Figure 5)

The yoke design consists of a flat steel plate with a machined center track. As the disk rotates, the bearing slides easily with the yoke. The yoke is connected to a vertical shaft by welds along the spliced end of the shaft (Figure 6).

The vertical shaft is guided by ball bushing linear pillow blocks. These blocks are bolted to a second vertical wooden member. The shaft is capable of linear motion only. Before the auger is attached rotational motion must be included. This is accomplished by coupling an intermediate shaft using a ball bearing coupler (Figure 7). the coupler will allow the lower shaft to rotate while oscillating vertically. The guides are suited to this purpose.

A hexagonal adaptor 1" in diameter is welded to the lower end of the shaft. The adaptor and auger are coupled with a pin connection. The auger extends down into a bin.

The stroke length of the auger is determined by the roller bearing position on the disk. A six inch stroke length corresponds to a bearing position three inches from the center of the disk. The disk is machined with six holes at 1/2" increments from the center of the disk starting at a half inch radius. In order to change the stroke length the roller bearing is simply unbolted and moved to a different hole.Thus, six different stroke lengths are available, the minimum stroke is one inch and the maximum stroke is six inches (FIgure 8).

The reciprocating motion is powered by a 1/3 horsepower constant speed DC motor. The motor drives the horizontal shaft attached to the disk through the use of a belt and pulley system. The speed of the motor is 1725 RPM. Thus, a reduction ratio of 17:1 is required to produce the desired output speed. This can be accomplished by coupling the motor shaft to a gear reduction mechanism.

A constant force spring is inserted along the vertical shaft under the yoke. The purpose of the spring is to counter the weight of the auger and reduce the power requirements of the system. A further discussion of counterbalancing appears later in the report.

The entire mechanical system is supported by a network of trusses made from angle irons. The wooden members are bolted to the angle irons while the angle irons are welded in place. The system is completely supported and portable (Figure 9).

The auger dirt removal system consists of three basic parts: the auger bin, auger tube, and auger spillway. These three pieces only serve the purpose of environment for testing the auger's performance, and do not contribute to the mechanical nature of the jig. All three pieces are made from translucent plastic so that visual inspection of the quality of operation can be monitored.

The auger bin is constructed entirely of plexiglas or Lexan, whichever is readily available. The joints and seams are bonded together with cement developed specifically for the plastic. The auger bin is able to hold eight cubic feet of dirt.

The auger tube is a clear tube of Lexan two feet long and has a wall thickness of 1/4". The tube will encase the auger so that when shavings are brought above ground level, they do not fall off the ramps. This will allow visual inspection and also allow the dirt to be redirected via the auger spillway so that the quantity of dirt removed can be measured.

An auger spillway is designed to allow the shavings brought through the auger tube to be redirected and displaced in an area which is separate from the auger bin.

Testing Procedure

Overview

Our testing program for the staircase auger will not focus on, or attempt to simulate the effects that the Moon's gravity and atmosphere will have on the operation of the auger since all calculations are based on the Earth's gravity. Previous groups have determined the optimum design characteristics of the auger. However, our auger has been designed and manufactured by an outside source. The helix angle of this auger is 25° versus a steep 60° for the auger modeled in previous 4182 design groups. Consequently, step height must be calculated for the auger based upon the mathematical model previously established. When the optimum step height is obtained, then testing can begin.

Methods of Observation

Several variables must be paid attention to in the design of the testing system. The design must be include the ability to monitor:

•Rotary speed.

•Reciprocating frequency.

•Dirt removal rate.

•Dynamic behavior of the cuttings.

A testing system was designed to observe all of the prior information. A great deal of time was devoted to designing a complete system which will faithfully reproduce the motions which are prescribed, so a method was developed to measure the system performance. In addition, we looked carefully at ways to look at the manner in which the cuttings were removed.

The rotary speed will be monitored by a commercially available hand-held tachometer. These are available in digital or analog readouts, and indirect or direct reading. The inexpensive models have direct contact with the spinning shaft. We suggest the least expensive, as they are sufficiently accurate for the purpose here. See Appendix 4 for example.

The rotary speed of the auger is the rotational speed of the vertical shaft, so speed can be directly monitored. The speed will vary according to the pulley ratio on the vertical shaft.

Reciprocating frequency will be monitored indirectly by using the hand-held tachometer on the horizontal shaft.. Knowing the relationship:

1 revolution of horizontal shaft = 1 cycle of vertical shaft

the frequency of the auger shaft can be easily determined.

Dirt removal rate is most easily monitored by weighing the dirt which has been lifted out of the auger test bin through the auger tube, and over the auger spillway during a period of time.

In addition to maintaining a record of the performance of the auger, visual data has been added to the list of necessary items to observe. The nature of the design depends on the behavior of the particle as it progresses from the bit to the top of the auger. By using a high speed camera system and a translucent drill string, we can watch the way that particles make their way up the ramp. From this information, we can look into and gauge the performance of the staircase auger relative to any changes made in speed, thrust, or material used.

Process

Once the auger is set-up and connected to the jig assembly, the auger tube should be attached to the auger in a manner consistent with the supplier's design. After the tube is in place, the auger spillway can be placed under it, and slipped up the auger. Now the auger bin is put under the auger, and the spillway lowered on top of the bin. This arrangement allows the test to be easily observed.

Drive Shaft

A force analysis was done on the drive shaft mechanism in order to determine the life of the shaft and pillow blocks. The calculations for this analysis are contained in the appendix. The following assumptions have been made:

•Assume tension on the drive pulley is negligible. This is the most severe case since any force in the y-direction on the pulley would tend to decrease the force acting on point A.

•The total weight acting on the drive disk equals 25 lbm.

•The maximum shaft speed is 100 rpm.

The calculated maximum radial force on the pillow block bearings was 28.5 lb. According to the data obtained on the Hub City PB100 pillow block bearing for a life of 50,000 hours and a moderate shock operating condition, the maximum radial load capacity was approximately 297 lbs. Therefore, the load on the pillow blocks is insignificant. The maximum deflection of the shaft was determined to be 7.2 x 10-5 in. This is obviously insignificant.

For the given loads and dimensions of our shaft the factor of safety for infinite life was calculated. Infinite life was chosen since the loads on the shaft were small coupled with a rather large shaft diameter (0.5 in.) It was inferred that the shaft would be over designed for finite life and thus infinite life was chosen. Our calculations show that for the given shaft parameters a factor of safety (n = 15) applied to the shaft. When a factor of safety of 2 was assumed. It was found that the shaft diameter could be 0.25 inches and still have infinite life. Due to the fact that pillow blocks with a 0.25" bore were not feasible, the overdesign using 0.5 inches was chosen.

Position & Inertial Force

The position of the yoke assembly is a sinusoidal function of the angular position of the roller bearing.

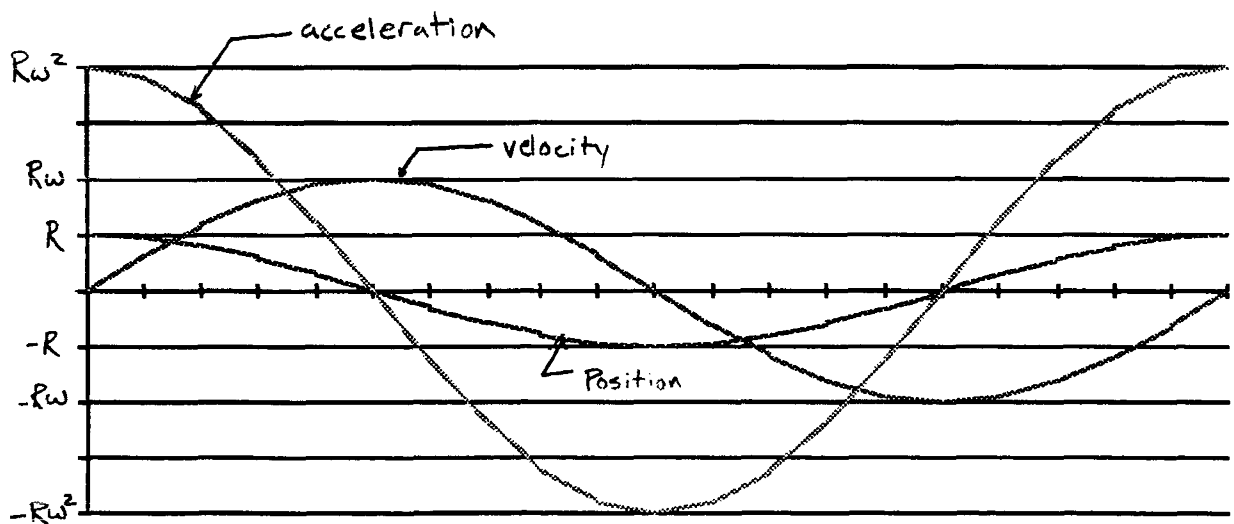
The position of the yoke assembly at angle O and roller bearing radius R is defined as :

$$X = R\cos O$$

Differentiating we can determine yoke velocity and acceleration. (w = angular velocity)

$$Velocity = -Rw\sin O$$

$$Acceleration = -Rw^2\cos O$$



Thus, a maximum acceleration of the yoke occurs at 0° and 180°. This is where the inertial force of the yoke is also a maximum. The inertial force is the product of mass and the inertial component of acceleration. An inherent jerking of the system takes place because of these inertial forces. In addition, a maximum thrust force is created at 180°.

Power Transmission

In order to determine the motor type and size that must be used to drive the reciprocating mechanical system, a power analysis must be performed. The analysis determines the maximum horsepower required and the relationship between speed and torque of the system. From this data a proper motor can be selected.

The horsepower required is based upon the torque needed to drive the disk at the given angular speed. (Power = Torque • Angular speed). The torque needed to drive the system is based upon:

- moment arm (distance from roller pin to center of disk)
- mass of assembly (yoke, auger and dirt)
- Acceleration of the assembly.
- Acceleration due to gravity.

The total acceleration of the system takes into account both the gravitational and rotation - induced acceleration.forces. The net acceleration is based on the angular position of the disk. The force due to the assembly on the roller is then mass times net acceleration. Horsepower at any given angle is determined by this torque and the angular velocity. Maximum required horsepower will correspond to the upper limits on the ranges of stroke length and reciprocation frequency. The upper limit of the stroke length is six inches and the frequency is 100 cycles per minute. The resulting maximum input horsepower required is less than 1/10 hp. However, parasitic losses will exist in the mechanical system due to the friction and imbalance, so we will arbitrarily set the design efficiency of the system at 20%. Therefore, actual horsepower needed to drive the mechanical system is 1/2 hp. (see calculations, Figure 10)

The above procedure only takes into account the force due to the yoke assembly acting on the roller. Simple counterbalance of the system will significantly decrease the required power input as well as help to alleviate the bucking of the assembly due to inertial forces.

The method of counterbalancing consists of a constant force spring acting up on the yoke assembly. The force in the spring should be equal to the weight of the yoke and auger. Thus, power required to overcome gravitational forces are canceled by the force from the spring. After counterbalancing and assuming 20% efficiency of the system the horsepower needed is 1/10 hp. (see Figure 10) The system will experience bucking regardless of the counterbalancing. However, the inertial forces generated will be slightly dampened by the spring.

Finally, the type of motor used must be considered. The ideal situation would be to use a variable speed dc motor hooked up directly to the disk. The characteristic speed curve of the motor should be above the characteristic speed curve of the system (Figure 11). However, the dc motor would require the use of a voltage regulator at a high added expense. Also, there is 1/3 hp constant speed ac motor available to the ME 4182 class. By using this motor considerable expense can be avoided.

In order to produce the rotational motion of the auger a second motor is necessary. The belt of the motor could be hooked directly to the shaft of the auger. The power needed to achieve the motion is minimal since the only opposing forces are the frictional effects of the bearings in the coupler and dirt on the auger. Thus almost any size motor will provide the necessary torque

Cost

Parts List			
Quantity	Description	Model	Cost
6'	2" x 8" wooden lumber	n/a	5.00
40'	1" x 1" angle iron	n/a	50.00
2	hardened and ground shaft		
	1060 steel 60 case	qs 1/2<24	28.00
2	fixed diameter linear bearing		
	pillow blocks	spb-8	56.46
2	malleable iron pillow block		
	with set screws	pb100 1/2"	100.00
1	sphere roller bearing	sb22202w33ff	22.83
1	single reduction worm gear		
	reducer	w300	400.00
1	2 foot plexiglas tube	n/a	50.00
1	1/4" x 8' x 4' plexiglas	n/a	75.00
1	constant force spring	n/a	75.00
1	yoke & disk couple	n/a	80.00
-	specified bolts	n/a	5.00

Conclusions & Recommendations

Conclusions

The stepper auger jig design has been successfully developed. We developed a testing system that will accommodate any auger design. Both rotational and reciprocating speeds can be independently varied so that the auger's performance can be scrutinized under any condition.

The stepper auger has been mathematically modeled to show that it will effectively displace dirt with an auger of specific design. An auger of a different design was specified for testing purposes. Since this design has a different helix angle and no steps, a new set of operating constraints will have to be calculated.

Recommendations

A general recommendation to any group expanding upon this project is to become proficient with a CAD system. We also suggest some more specific ideas:

•Rework the program methodology which references the mathematical model presented by the Summer 1987 Auger group. Determine the necessary step height, and speeds for the auger.

•Design a gear reduction system in order to alleviate the use of an expensive manufactured gear box.

•Design a different method of counter balancing to alleviate the high cost of manufacturing a constant force spring.

•Utilize your time wisely and plan ahead - thoughts take time to mature. Delegate responsibility and set weekly goals.

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McMurray, Gary; Graduate Teaching Assistant; Georgia Institute of Technology

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Appendix #1 *Figures*

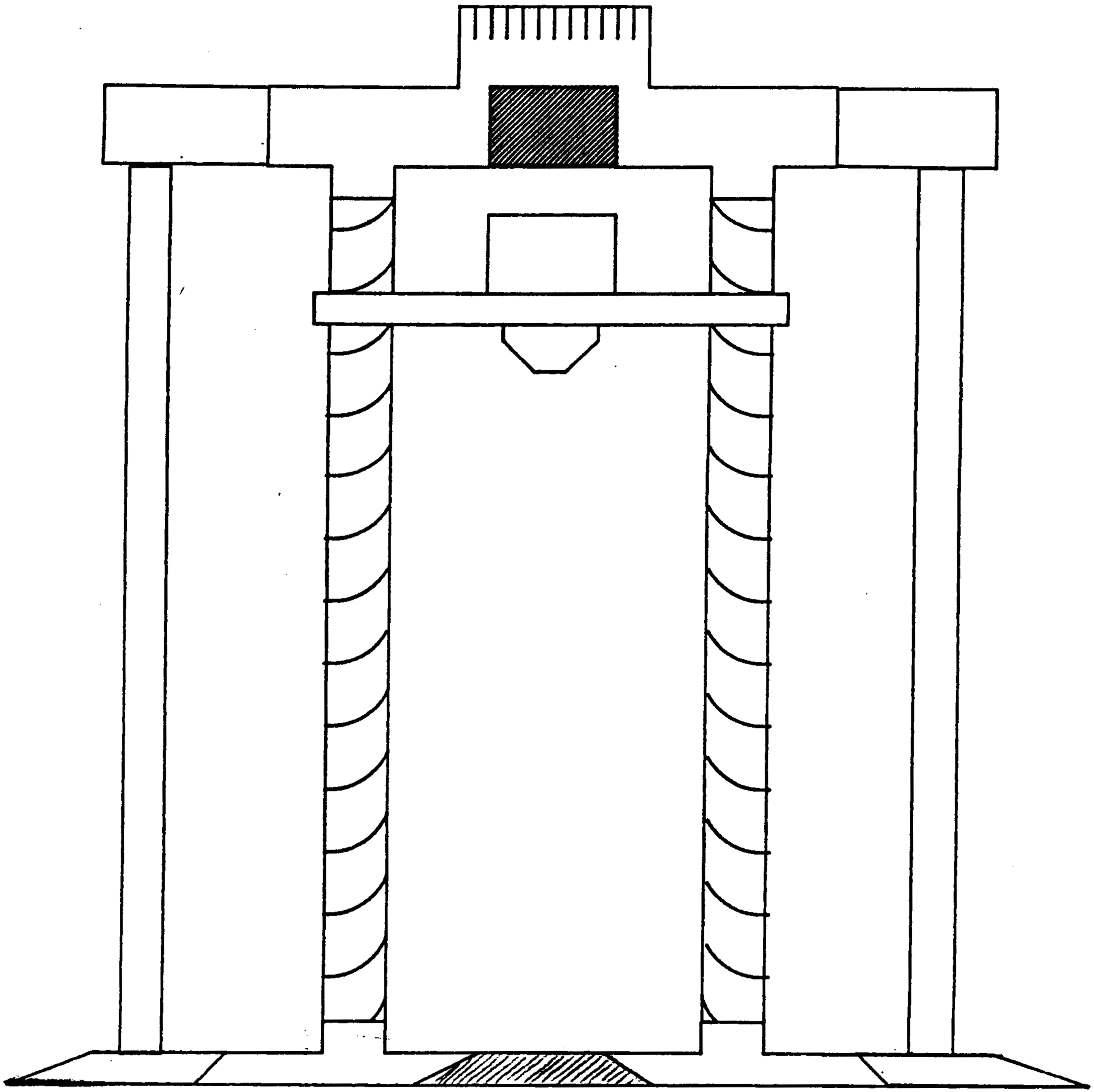


Figure 1

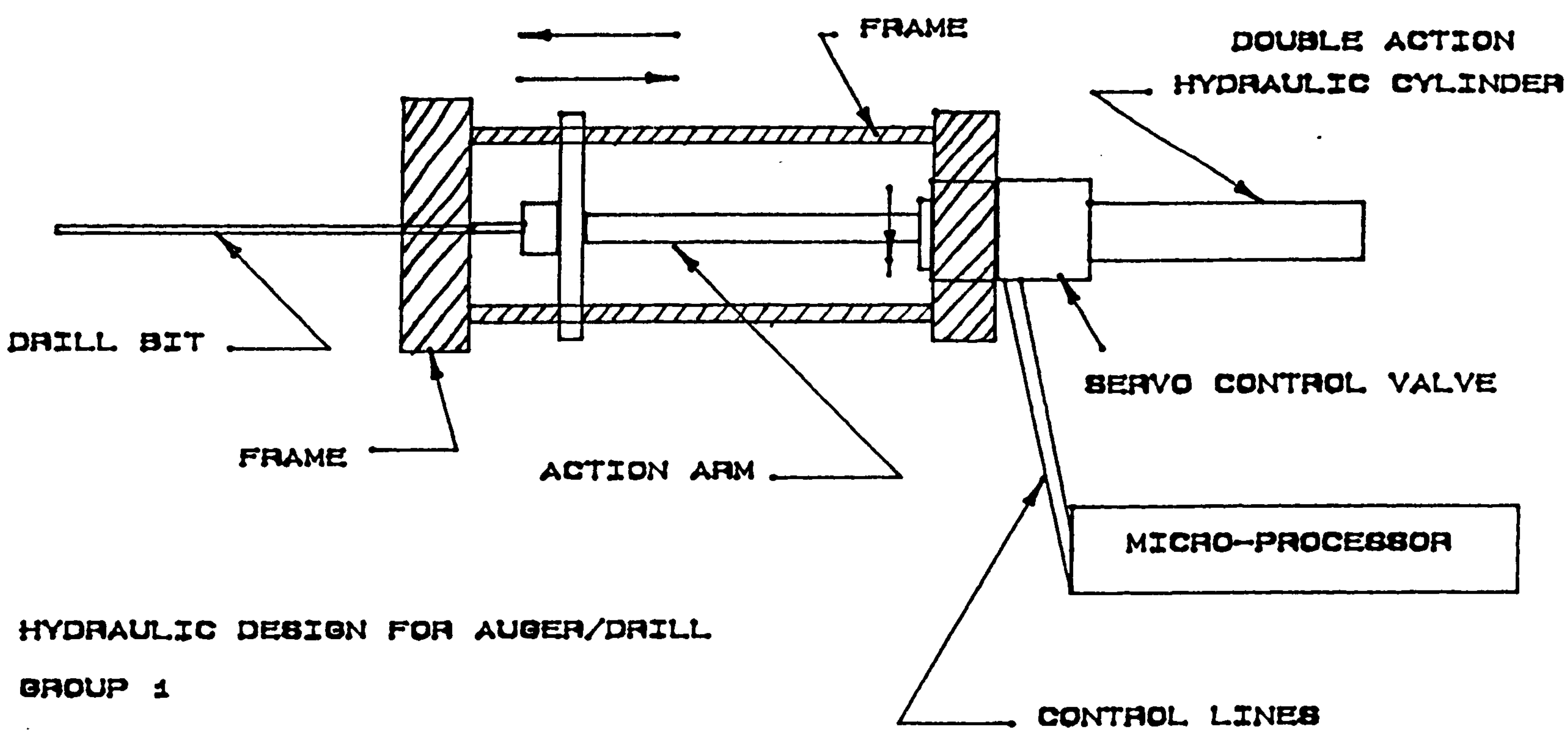

FRAME
DOUBLE ACTION
HYDRAULIC CYLINDER
DRILL BIT
SERVO CONTROL VALVE
FRAME
ACTION ARM
MICRO-PROCESSOR
HYDRAULIC DESIGN FOR AUGER/DRILL
GROUP 1
CONTROL LINES

Figure 2

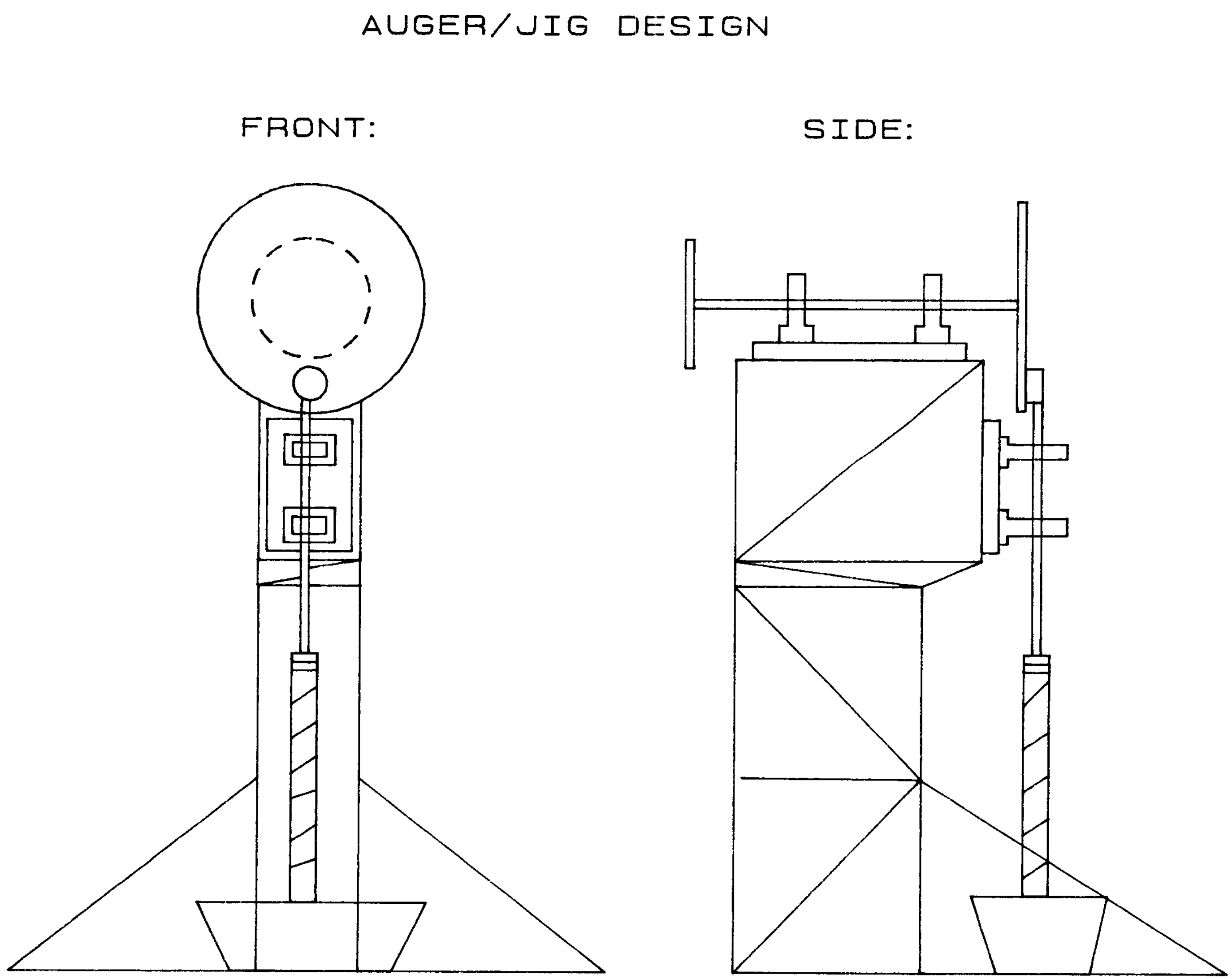


Figure 3

DRIVE SHAFT ASSEMBLY

A – WHEEL
B – SHAFT
C – PILLOW BLOCK
(PB100 1/2" SHAFT DIA, HUB CITY)
D – PULLEY
E – FRAME
F – RUBBER BUSHING
(1/2" THICK)

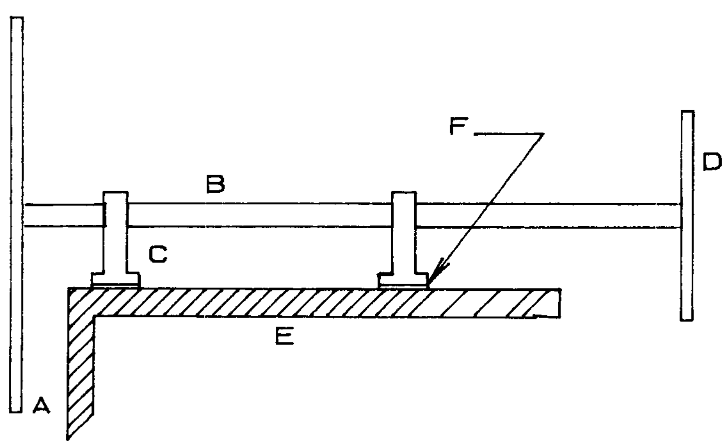
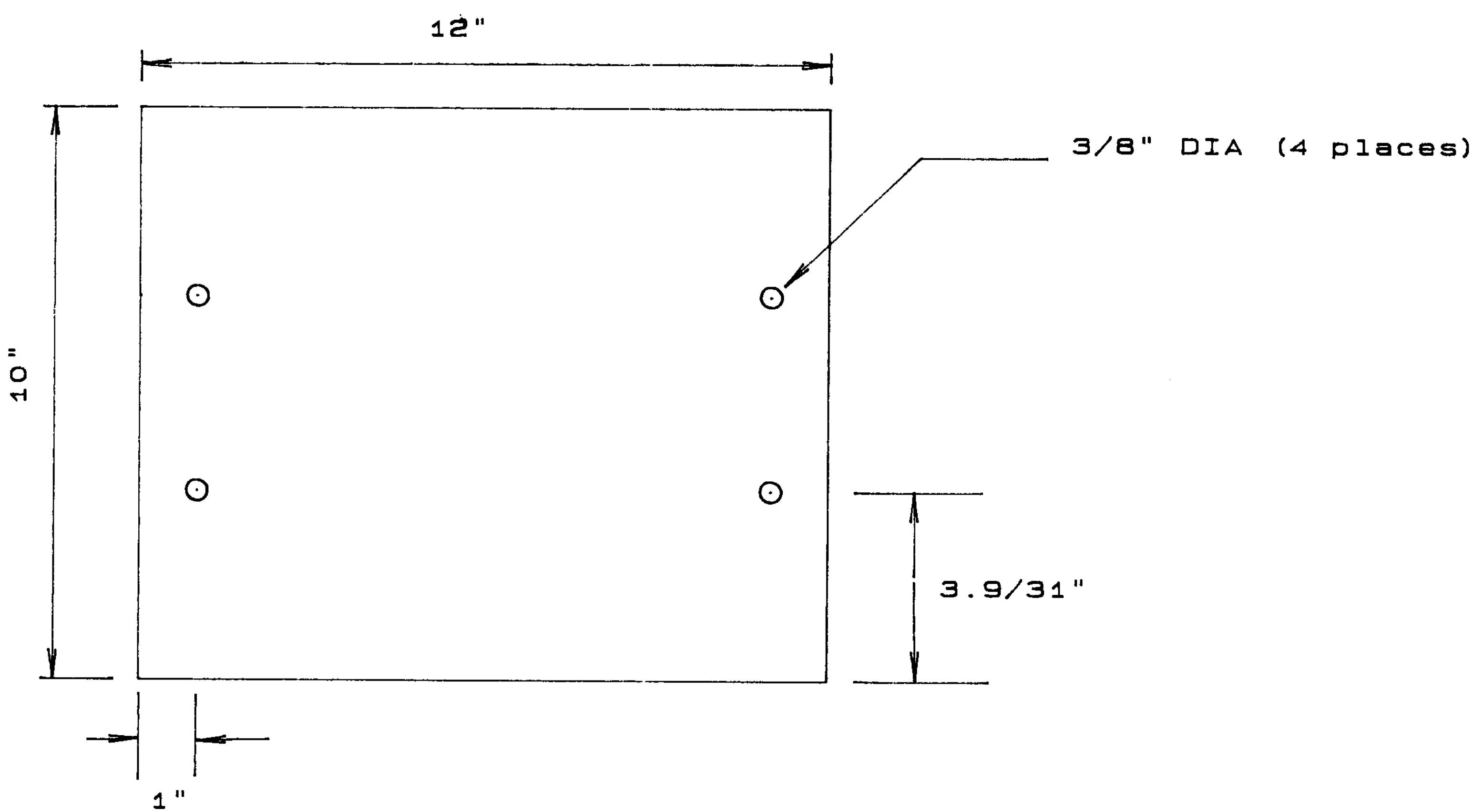


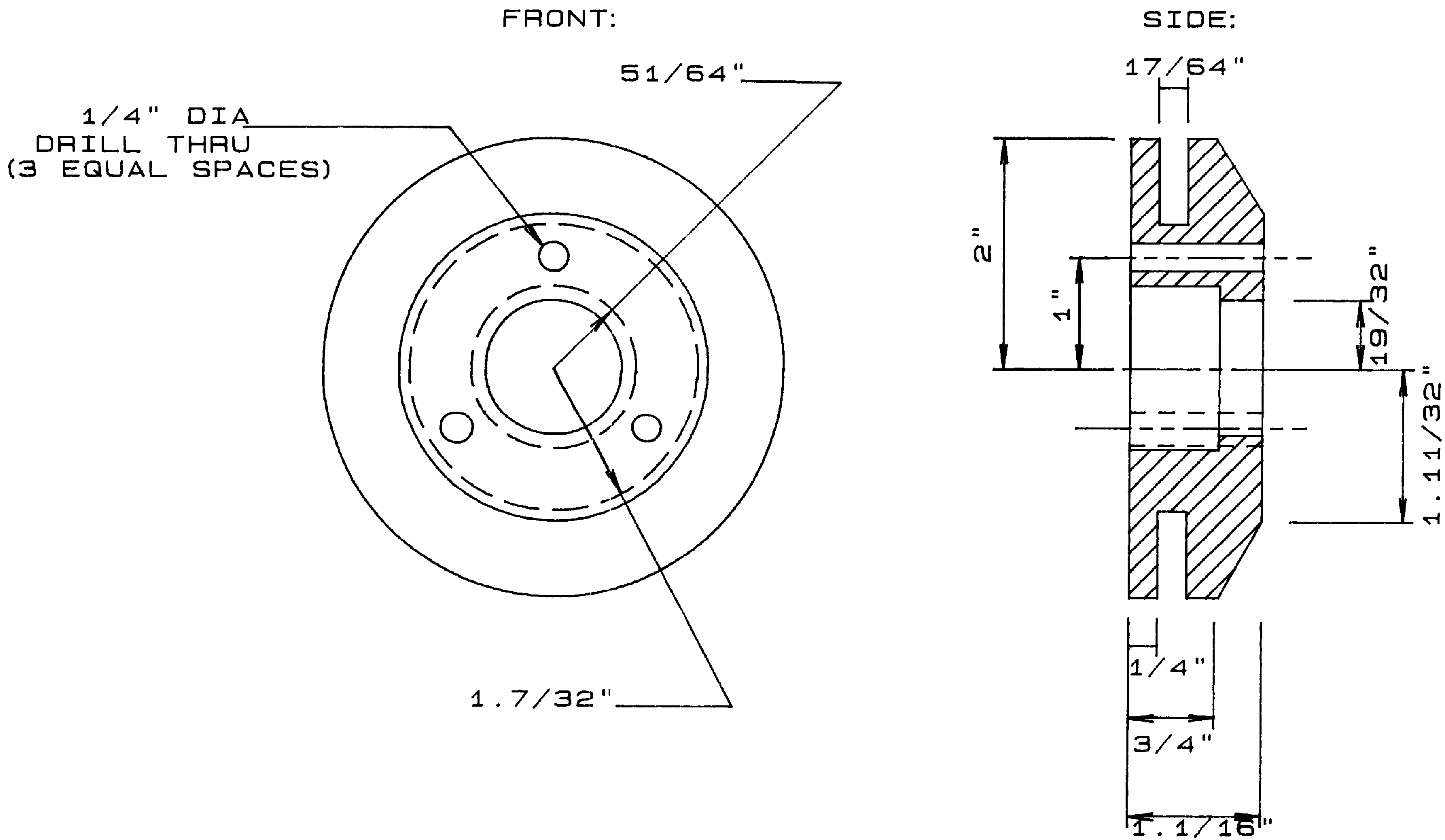
Figure 4a



TOP MOUNTING PLATE

NOTE: Material 1"x 10" oak wood

Figure 4b

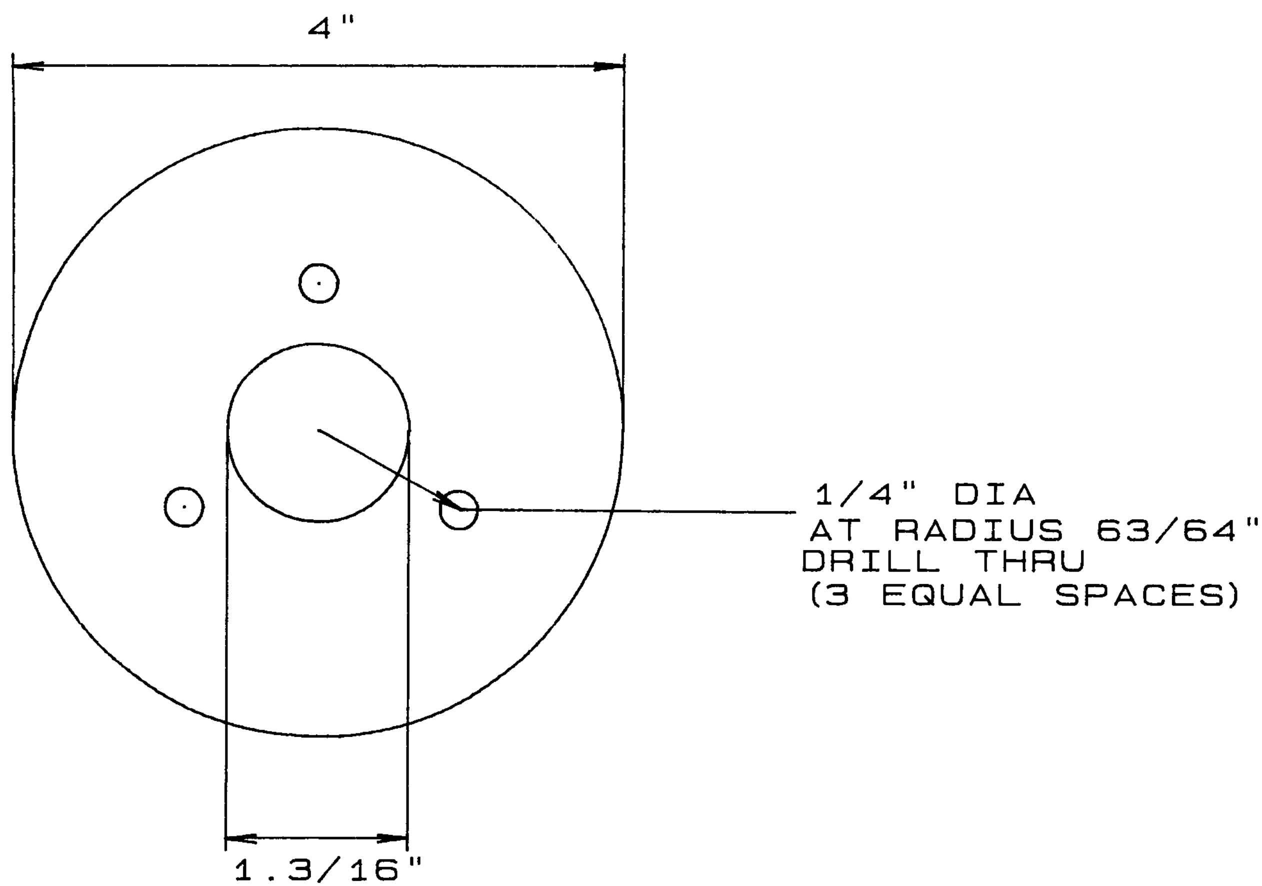


FRONT PLATE OF ROLLER

NOTE: All tolerances +- .002

Figure 5a

BACK PLATE OF ROLLER



NOTE: Material 1/4" 1050cd steel

Figure 5b

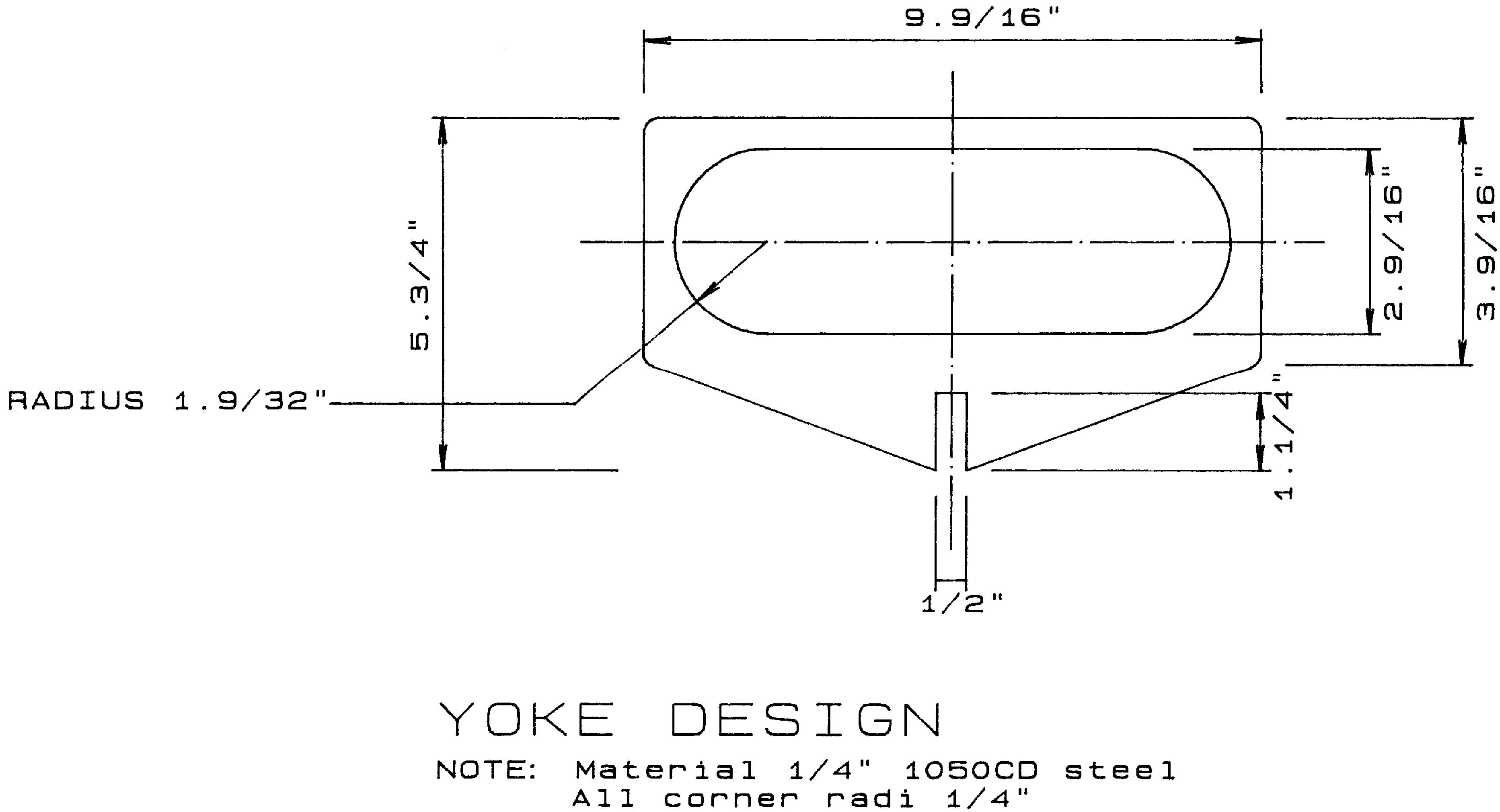

9.9/16"
5.3/4"
2.9/16"
3.9/16"
RADIUS 1.9/32"
1.1/4"
1/2"
YOKE DESIGN
NOTE: Material 1/4" 1050CD steel
All corner radi 1/4"

Figure 6

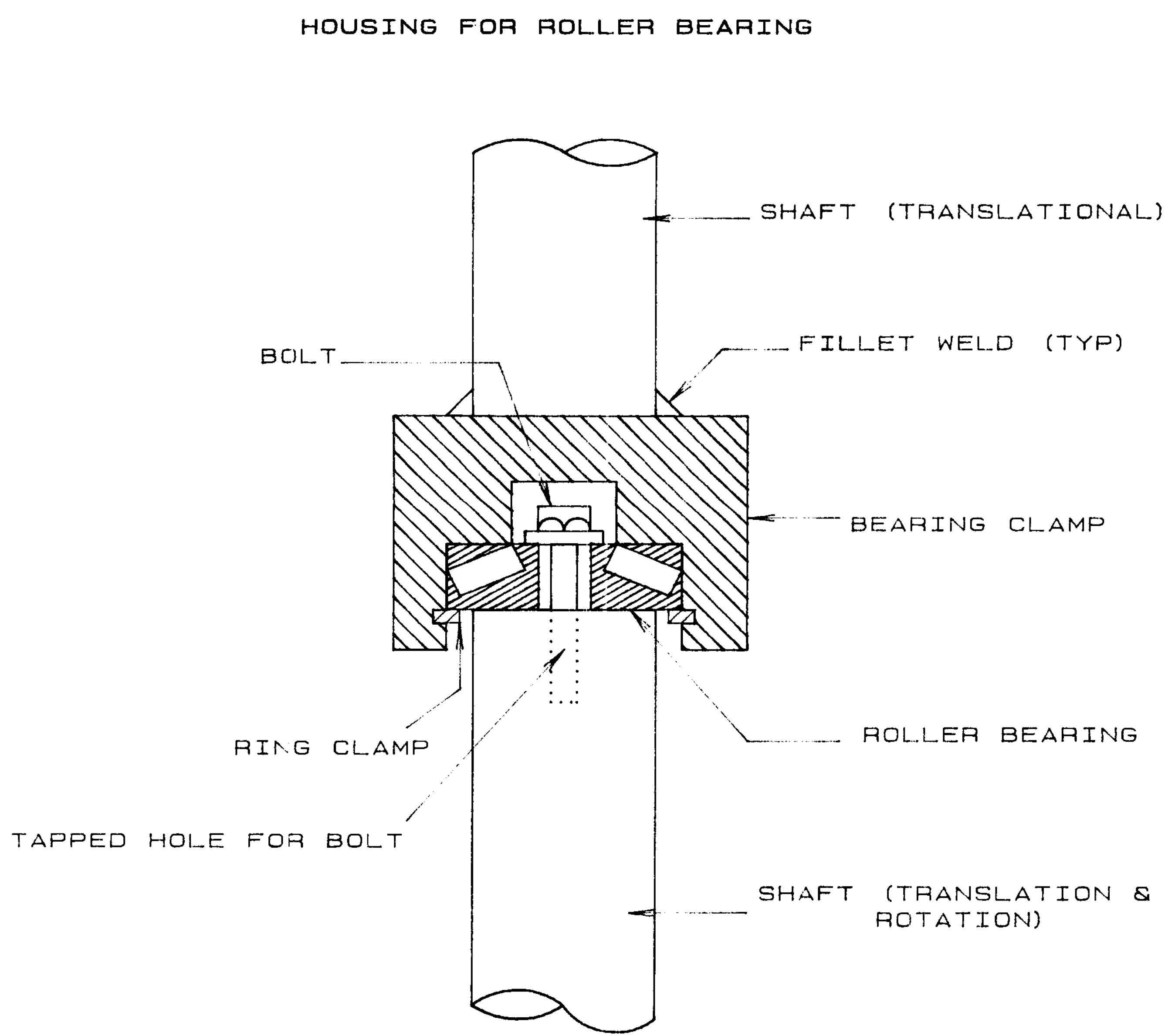
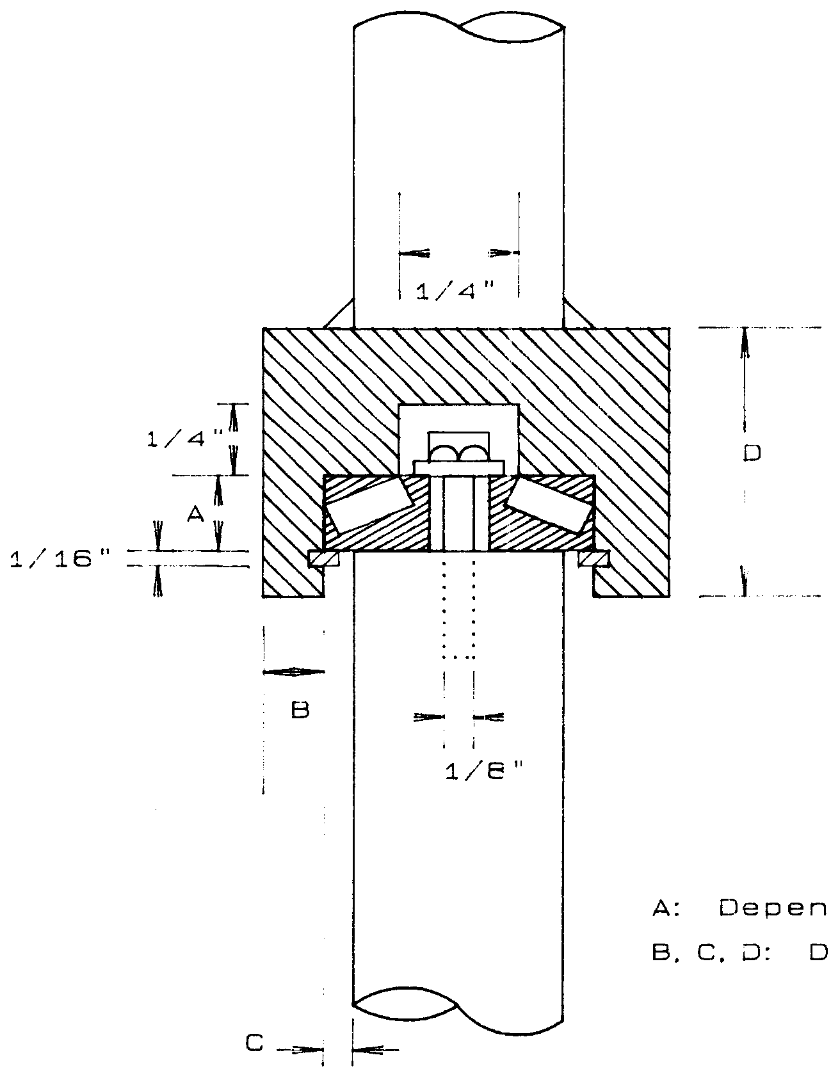

HOUSING FOR ROLLER BEARING
SHAFT (TRANSLATIONAL)
BOLT
FILLET WELD (TYP)
BEARING CLAMP
RING CLAMP
ROLLER BEARING
TAPPED HOLE FOR BOLT
SHAFT (TRANSLATION &
ROTATION)

Figure 7a

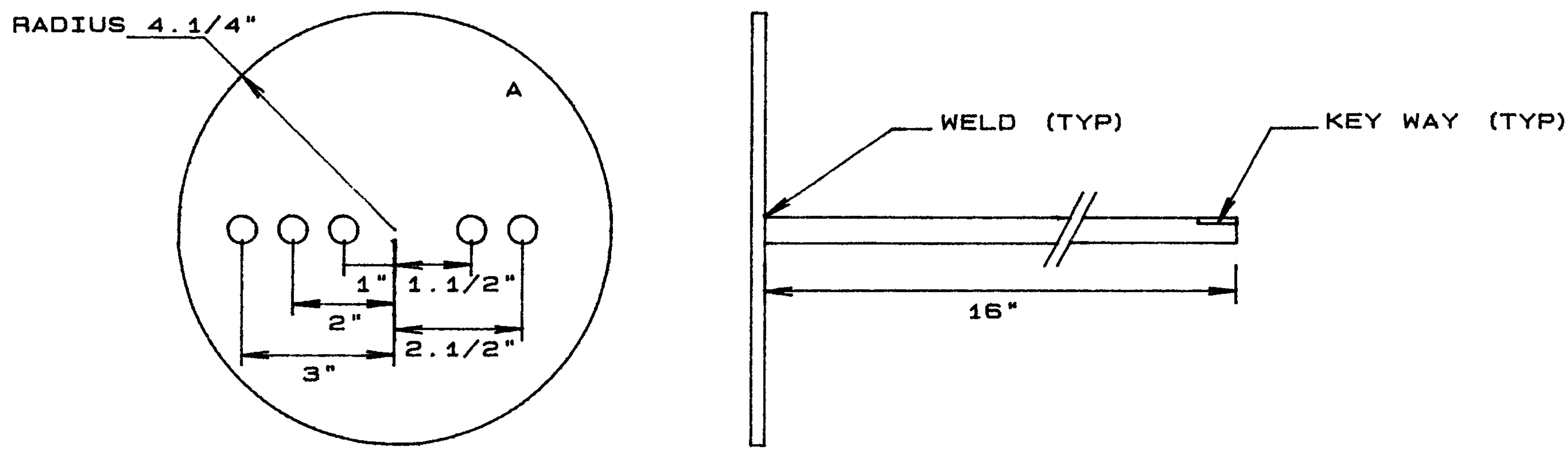
DIMENSIONS OF BEARING HOUSING:



A: Depends on height of roller bearing

B, C, D: Dimensions can vary

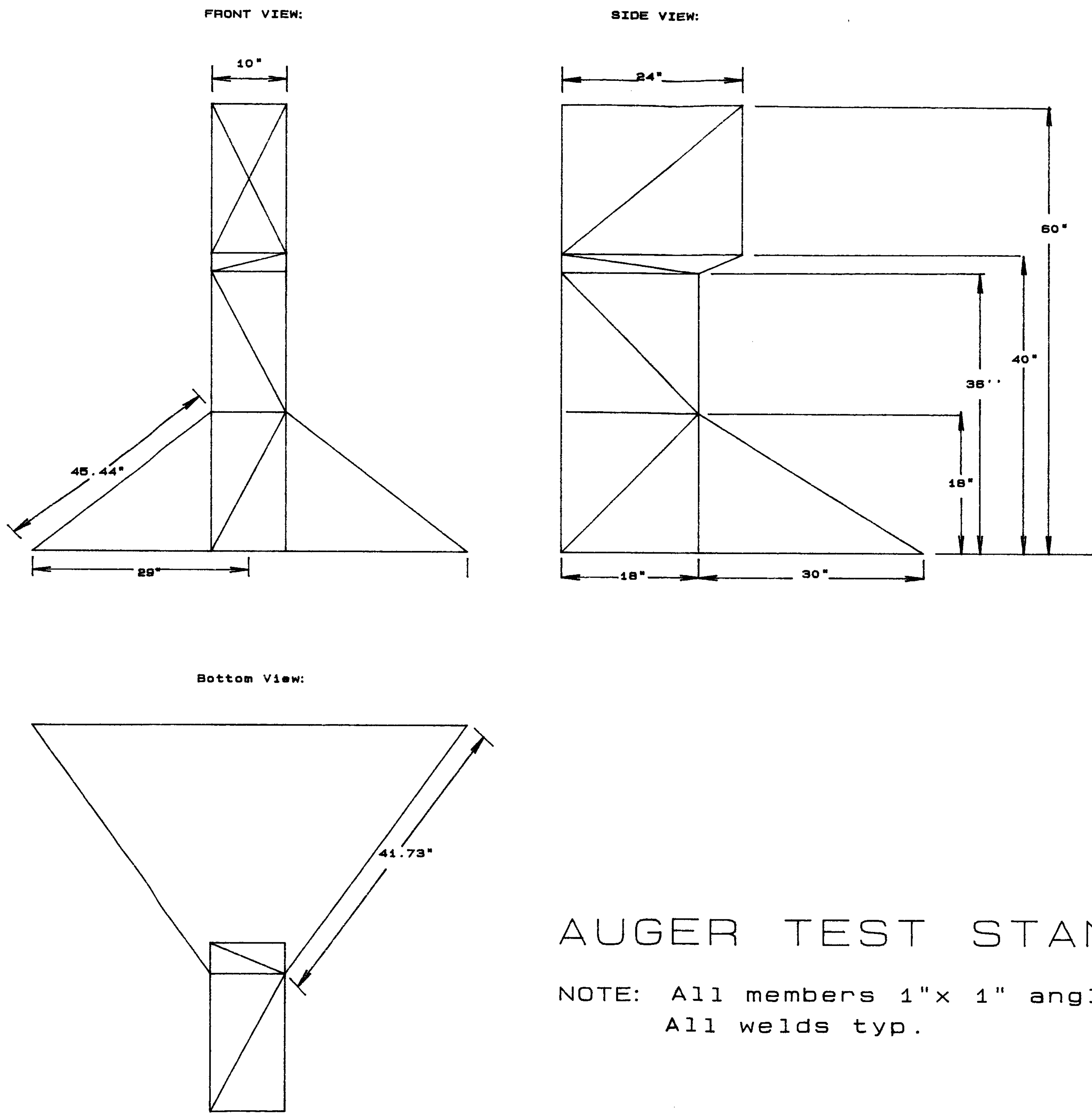
Figure 7b



DRIVER DISK & SHAFT DESIGN

NOTE: Disk material 1/4" 1050CD steel
Shaft material 1/2" DIA 60 Case (THOMSON)

Figure 8



AUGER TEST STAND

NOTE: All members 1"x 1" angle iron
All welds typ.

Figure 9

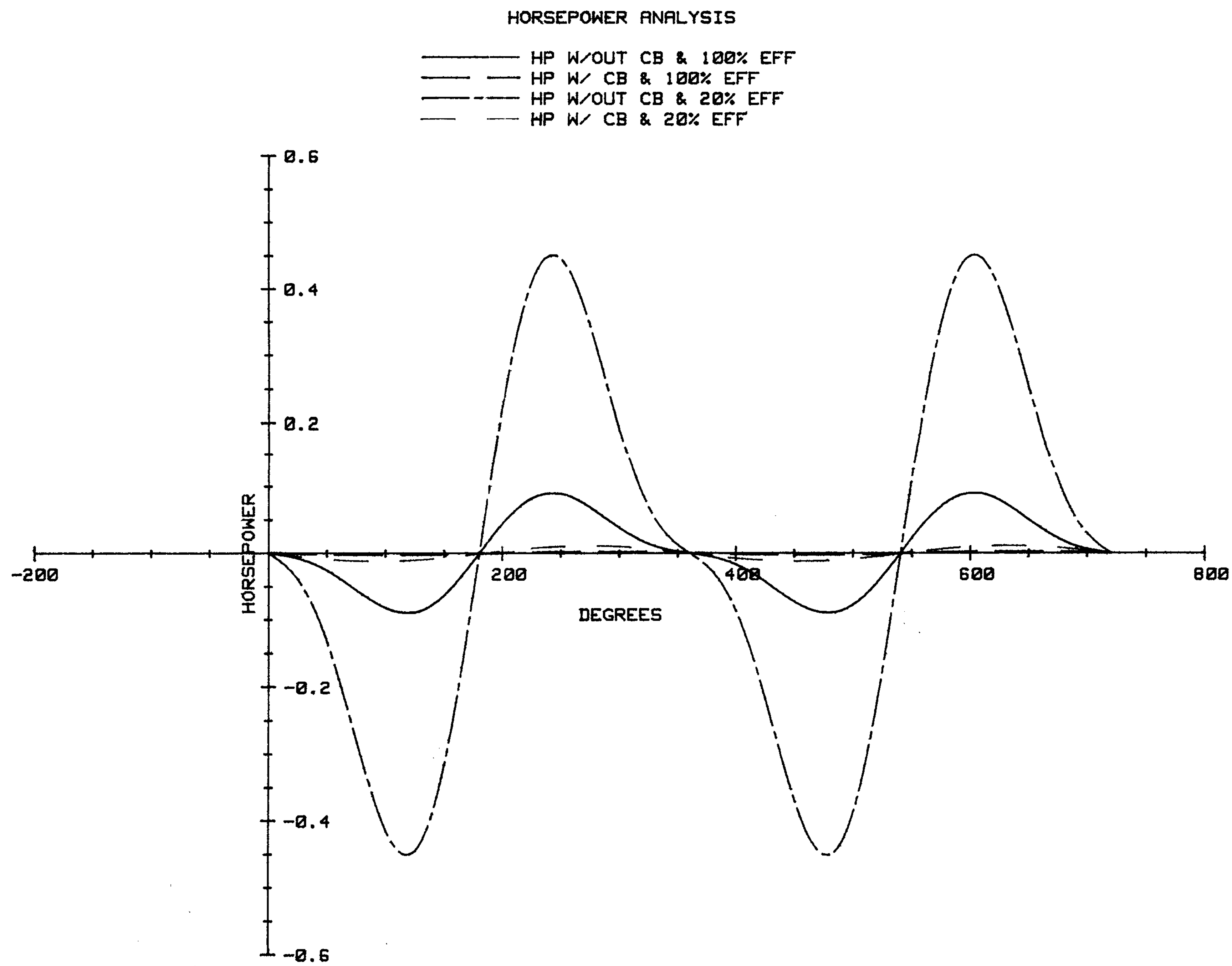
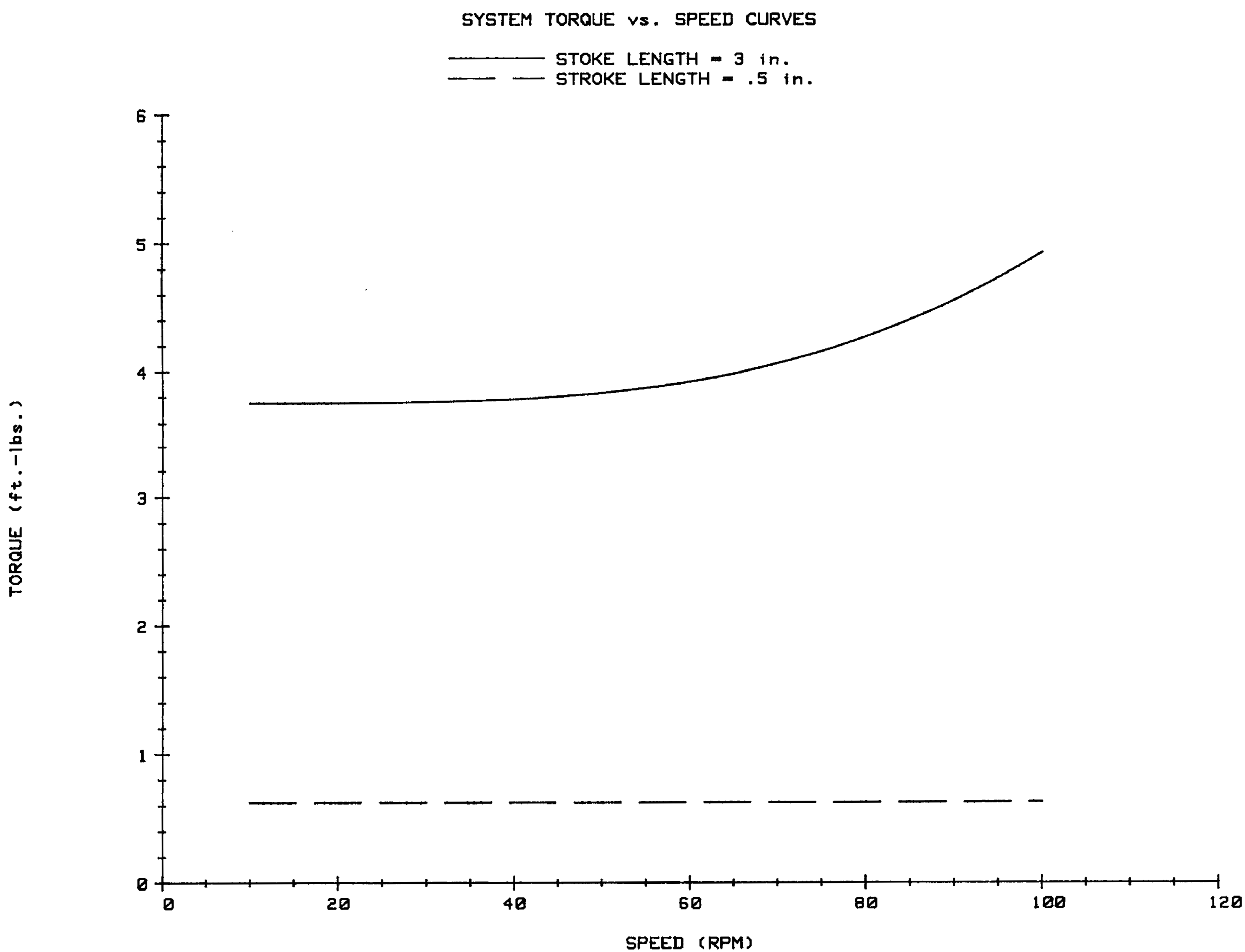

HORSEPOWER ANALYSIS
HP W/OUT CB & 100% EFF
HP W/ CB & 100% EFF
HP W/OUT CB & 20% EFF
HP W/ CB & 20% EFF
0.6
0.4
0.2
-0.2
-0.4
-0.6
HORSEPOWER
-200
200
400
600
800
DEGREES

Figure 10



SYSTEM TORQUE vs. SPEED CURVES
STOKE LENGTH = 3 in.
STROKE LENGTH = .5 in.
TORQUE (ft.-lbs.)
0
1
2
3
4
5
6
0
20
40
60
80
100
120
SPEED (RPM)

Figure 11

Appendix #2 *Calculations*

FORCE ANALYSIS :

During rotation of the disk the yoke is experiencing a change in velocity and acceleration according to the sinusoidal model;

$$x = R\cos\theta$$

then $V = -Rw\sin\theta$

and $A = -Rw^2\cos\theta$

where x = vertical position

R = radius of rotating disk

θ = Angular position of disk

This acceleration of the auger produces an inertial force which acts on the pin connected roller bearing. The pin is also supporting the weight of the auger. The force is :

$F = mg$ where m = mass

g = gravity

Thus the net force is :

$$F = mRw^2\cos\theta + mG$$

This force always acts in the direction of gravity. The force is maximum at top dead center or at $\theta = 0°$. It is minimum at bottom dead center or $\theta = 180°$.

Constraints for this force are:

$f = 100$ rpm or $\omega = 10.47$ rad/sec

Stroke length = 6 in

So $R = 3$ in or $.25$ ft

$\therefore \quad F = (25\ lb_m)(.25\ ft)(10.47\ \text{rad/sec})^2 \cos\theta + 25\ lb_m$

$F_{max} = 21.41 + 25 = 46.41\ lb_f$ ← excessive jerking of the system

$F_{min} = 0 + 25 = 25$ ← minimal jerking

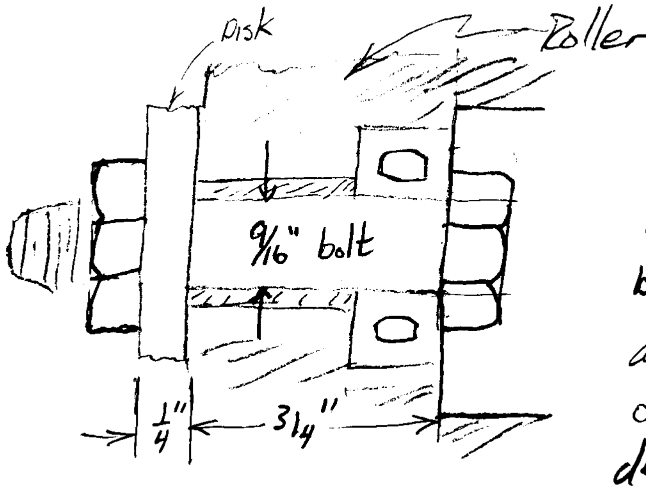
By using a 25 lb_f constant force spring acting to oppose the weight of the yoke. We can minimize the force acting on the pin. Thus the maximum and minimum forces are:

$$F_{max} = 21.41\ lb_f$$

$$F_{min} = 0\ lb_f$$

Thus, only slight jerking motion is present.

Bolt Analysis for Yoke Roller



*Note: The calculation of bending stress in bolt given by: $\sigma = \frac{Mc}{I}$ is an assumption due to the lack of knowledge of load distribution and deformations of bolt.[1]

FAILURE BY PURE SHEAR:

stress on bolt $\Rightarrow \tau_{bolt} = \frac{F}{A}$

where, F = maximum force (see Force Analysis on Pillow Blocks) ($F_{max} = 21.3$ lbf)
Assume factor of safety of 2

$F_{max} = 42.6$ lbf

$$\tau_{bolt} = \frac{42.6 \text{ lbf}}{\pi\left(\frac{9}{32}\right)^2} = \boxed{171.4 \text{ psi}}$$

Endurance Limit: $S_e = k_a k_b k_c k_d k_e k_f S_e'$ (see shaft analysis)

$S_e' = 97.5$ kpsi
$k_a = 1$
$k_b = 0.869\,(9/16)^{-.097} = 0.934$
$k_c = 0.814$
$k_d = 1$
$k_e = 1$

$S_e = (1)(.934)(.814)(1)(1)(97.5 \text{ kpsi})$

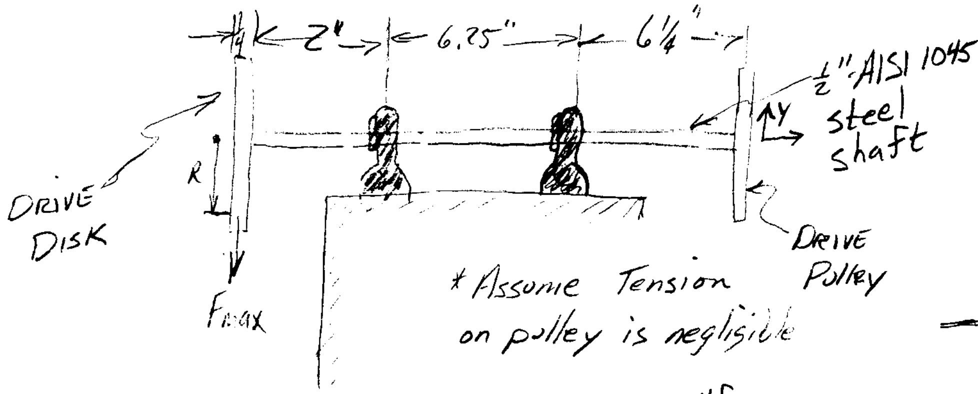
$\boxed{S_e = 74.2 \text{ Kpsi}}$

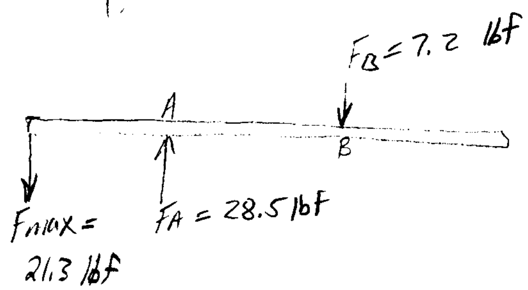
$\therefore \tau_{bolt}$ is allowable.

---

(1) Shigley, Mechanical Engineering Design, p 392, The author suggests, for design purposes, simply to increase the factor of safety.

MAXIMUM RADIAL FORCE ON PILLOW BLOCK BEARINGS





∴ The maximum radial force on the pillow block bearing will be $F_A = 28.5$ lbf.

FOR HUB CITY PILLOW BLOCK Bearings 1/2" PB 100 series: ①

- @ 100 rpm and steady load with Avg. life of 2500 hrs

  Radial Load Capacity = 1,150 lb

- for an average like of 50,000 hrs with moderate shock:

  Radial Load Capacity = (0.258)(1,150 lb)

  RLC = 296.7 lb

∴ The load on the pillow blocks are insignificant.

Acceleration

$$A = -R\omega^2\cos\theta$$

$$A_{max} = -R_{max}(\omega_{max})^2\cos(0°, 180°)$$

$$A_{max} = -(3.0\text{ in})(10.47\tfrac{r}{s})^2(1)$$

$$= 328.9\ \tfrac{in}{s^2}\cdot\tfrac{ft}{12\text{ in}}$$

$$= 27.4\ ft/s^2$$

$$\omega_{max} = 100\ \frac{rev}{min}\ \frac{2\pi\ rad}{rev}\cdot\frac{min}{60\,sec}$$

$$= 10.47\ \frac{rad}{s}$$

$R_{max} = 3$ in

$m_{max} = 25$ lbm

FORCE

$$F = ma$$

$$F_{max} = \left[\frac{25\text{ lbm}}{32.2\ \frac{lbm\cdot ft}{lbf\ s^2}}\right]\left[27.4\ \frac{ft}{s^2}\right]$$

$$\boxed{F_{max} = 21.3\text{ lbf}}$$

① $\sum F_y = 0 = F_{max} + F_B - F_A$

② $\sum M_A = 0 = F_{max}\left(\frac{2.125}{12}ft\right) - F_B\left(\frac{6.25}{12}ft\right)$

③ $\sum M_B = 0 = F_{max}\left(\frac{8.375}{12}ft\right) - F_A\left(\frac{6.25}{12}ft\right)$

① $F_A = F_B + F_{max}$

② $F_{max}\left(\frac{8.375}{12}ft\right) = (F_B + F_{max})\left(\frac{6.25}{12}ft\right)$

$$F_B = \frac{F_{max}\left(\frac{8.375-6.25}{12}\right)}{6.25/12\ ft}$$

$$\boxed{F_B = 0.34F_{max} = 7.2\text{ lbf}}$$

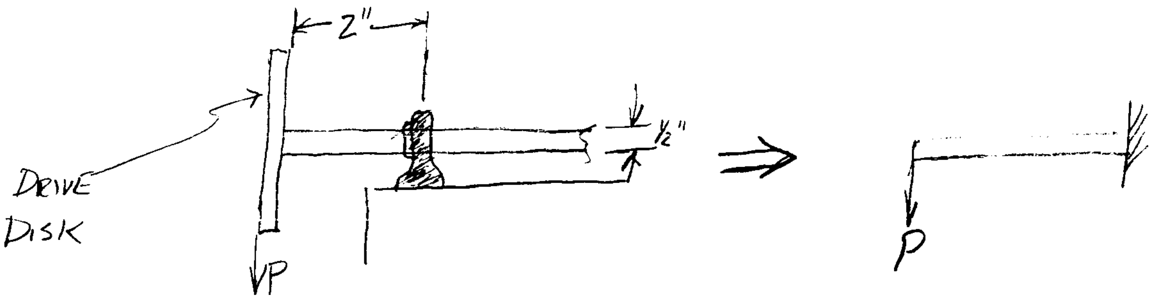
$$\boxed{F_A = F_{max} + F_B = 28.5\text{ lbf}}$$

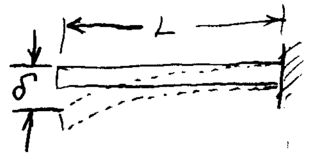
CHECK

③ $F_{max}\left(\frac{2.125}{12}ft\right) - F_B\left(\frac{6.25}{12}ft\right) = 0$

$45.25 - 45.25 = 0$ ✓

MAXIMUM DEFLECTION OF DRIVE SHAFT





$L = 2.125 \text{ in}$ $\qquad I_{ROD} = 0.017 \text{ in}^4$

$P = 25 \text{ lbm}$

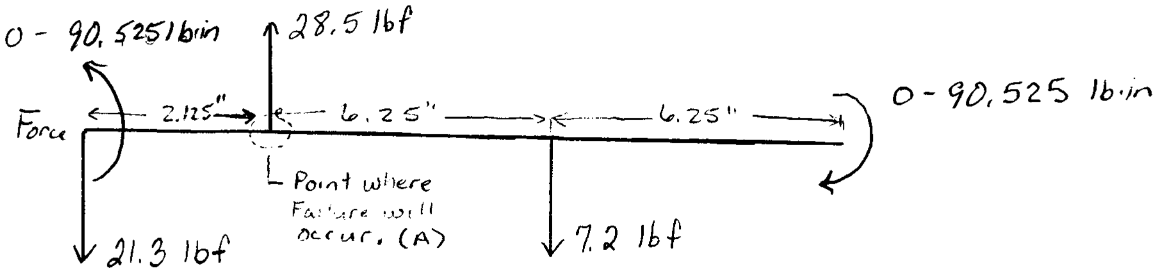
$$\delta_{max} = \frac{PL^3}{3EI} = \frac{(25 \text{ lbm})(2.125 \text{ in})^3}{3(30 \times 10^6 \text{ PSI})(0.017 \text{ in}^4)} = \underline{1.57 \times 10^{-4} \text{ in}}$$

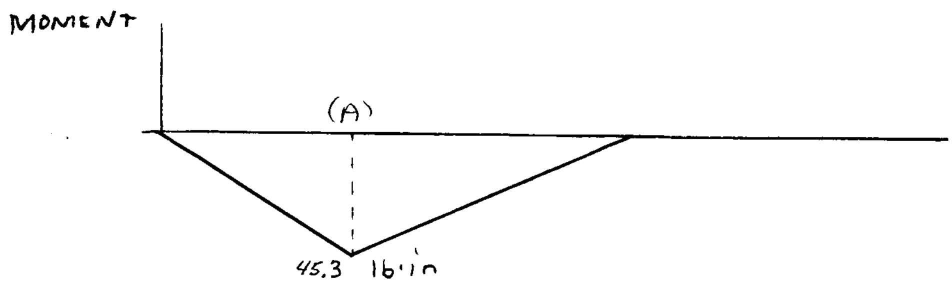
∴ The deflection of the shaft due to the maximum force from drive disk is insignificant.

---

① Specifications for the Hub City Pillow Block was obtained from: HUB CITY - ENGINEERING MANUAL, p411.

SHAFT ENDURANCE LIMT





ENDURANCE LIMIT

$\boxed{S_e = k_a k_b k_c k_d k_e k_f S_e'}$ (P. 287)

Using as our material AISI 1045 Q & T 720° F : (P. 823 SHIGLEY)

$S_e' = 0.5\ S_{ut} = 0.5(195\ \text{kpsi})$

$\boxed{S_e' = 97.5\ \text{kpsi}}$

MODIFYING FACTORS :

Surface finish: (assuming polished steel) (P. 288)

$\boxed{k_a = 1}$

Size effect :

$k_b = 0.869\, d^{-0.097}$; $0.3" < d \leq 10"$ (P. 293)

$k_b = 0.869(.5")^{-0.097}$

$\boxed{k_b = 0.92}$

Reliability : (assume we want 99% reliability) (P. 299)

$\boxed{k_c = 0.814}$

Temperature Effects :

$\boxed{k_d = 1}$; $T \leq 450°C$ (P. 300)

Stress Concentration :

Since there are no notches or keys at the point where failure is most likely to occur take :

$\boxed{k_e = 1}$

Therefore;

$$Se = (1)(0.92)(0.814)(1)(1)(97.5\ kpsi)$$

$$\boxed{Se = 73\ kpsi}$$

For our design and the given loads the factor of safety employed for our shaft can be found using the equation:

$$d = \left\{ \frac{32(n)}{\pi} \left[\left(\frac{T}{S_y}\right)^2 + \left(\frac{M}{Se}\right)^2 \right]^{1/2} \right\}^{1/3} \qquad (1)$$

where; d = shaft diameter

T = max Torque at point of failure

M = max moment at point of failure

S_y = yield strength of shaft

Se = endurance limit.

From the diagrams for Force and Moment

T = 90.525 lb·in

M = 45.3 lb·in

S_y = 185 kpsi

Se = 73 kpsi

d = .5 in

Substituting into (1) this gives

$$(0.5) = \left\{ \frac{32(n)}{\pi} \left[\left(\frac{90.525\ lb\cdot in}{185\ kpsi}\right)^2 + \left(\frac{45.3\ lb\cdot in}{73\ kpsi}\right)^2 \right]^{1/2} \right\}^{1/3}$$

which gives:

$$n = \frac{.01227}{\left[\left(\frac{90.525}{185\ k}\right)^2 + \left(\frac{45.3}{73\ k}\right)^2 \right]^{1/2}}$$

$$n = 15.5$$

$$n = 15$$

SPRING DESIGN CALCULATIONS:

CONSTANT FORCE SPRING CALCULATIONS -

(1) $P = \dfrac{E b t^3}{26.4 R_n^2}$ lbs (2) $S = \dfrac{E t}{2 R_n}$ psi

(3) $R_2 = 1.15 R_n$ (4) $S_f = t/R_n$

$S_f = 0.010$ for 50,000 cycles

$S_f = 0.025$ for 5000 cycles

Carbon steel : $E = 30 \times 10^6$ psi

stainless steel : $E = 28 \times 10^6$ psi

PARAMETERS:

$P = 25$ lbs.

R_2 = Storage Drum Radius

R_n = Natural Radius

S = stress

b = Material Width

t = Material Thickness

* Rated load reached after initial deflection of 1.25 x OD

DESIGN CONSTRAINTS:

$R_n \geq 1''$

$0.015 \leq S_f \leq 0.010$

$\left.\begin{array}{l} \text{spring Length} = 15\text{ in} \\ R_n = 4\text{ in} \\ t = 0.06\text{ in} \\ b = 1.304\text{ in} \\ S_f = 0.015\text{ kpsi} \end{array}\right\}$ Final Design

Appendix #3 *Programs*

```
1       REM************************************************
2       REM THIS PROGRAM CALCULATES HORSEPOWER REQUIREMENTS FOR OUR MECHANICAL SYS
3       REM R=MAX MOMENT ARM(HALF OF STROKE LENGTH)
4       REM CPM=CYCLES PER MINUTE;W=ANGULAR VELOCITY
5       REM G = ACCEL. OF GRAVITY; M = MASS OF AUGER AND YOKE
6       REM MP = MASS OF DIRT PARTICLES; EFF = MECHANICAL EFFICIENCY
7       REM************************************************
8       REM INITIALIZE VARIABLES
10      R=.25
20      Cpm=100.
30      W=(Cpm*PI)/30
40      G=32.2
50      M=15./G
60      Mp=.5/G
70      Eff=1
71      REM************************************************
72      REM CREATE FILES FOR STORING DATA
80      CREATE ASCII "CHP1",80
90      ASSIGN @Path_1 TO "CHP1"
100     CREATE ASCII "HP1",80
110     ASSIGN @Stroke TO "HP1"
111     REM************************************************
112     REM SET UP A LOOP TO ITERATE POSITION ANGLE THETA
113     REM WHILE CACULATING P (MAX POWER); CP(MAX POW. W/COUNTERBALANCING)
114     REM AND HP
120     FOR Theta=0 TO 720 STEP 5
130     Phi=Theta*PI/180
140     P=1/Eff*(W*(M*G*R*COS(Phi+PI/2))+M*(R^2*W^3*COS(Phi)*SIN(Phi))-Mp*G*R*W*SI
N(Phi))
150     Cp=1/Eff*(-Mp*G*R*W*SIN(Phi))
160     Hp=P/550
170     Chp=Cp/550
171     REM************************************************
172     REM SEND DATA TO OUPUT FILES
180     OUTPUT @Path_1;Theta," ",Chp
190     PRINT "CHP=";Chp," ","THETA=";Theta
200     OUTPUT @Stroke;Theta," ",Hp
210     PRINT "HP=";Hp," ","THETA=";Theta
220     NEXT Theta
230     ASSIGN @Path_1 TO *
240     ASSIGN @Stroke TO *
250     END
```

```
10     REM************************************************
20     REM MAX/MIN TORQUE SPEED CURVES
30     REM WEIGHT = F =15 LBS.
40     REM RPM = REV PER MIN
50     REM R = STROKE LENGTH OF MECHANISM IN INCHES
60     REM W = ANGULAR VELOCITY
61     REM M= MASS OF YOKE PLUS AUGER
63     REM G= ACCEL. DUE TO GRAVITY
70     REM************************************************
71     G=32.2
80     R=.041667
90     M=15./G
91     REM************************************************
92     REM CREATE A FILE TO STORE TORQUE DATA
94     CREATE ASCII "TORQ2",80
95     ASSIGN @Stroke TO "TORQ2"
96     REM************************************************
97     REM SET UP A LOOP TO ITERATE RPM
99     FOR Rpm=10 TO 100 STEP 10
100    Amax=0.
101    W=Rpm*PI/30
102    REM************************************************
103    REM SET UP A NESTED LOOP TO ITERATE POSITION ANGLE THETA
105    FOR Theta=0 TO 360
106    REM CALCULATE ACCELERATION OF PIN ON DISK THEN DETERMINE MAX ACCEL.
108    A=-R*W^2*COS(Theta)-G*SIN(Theta)
109    IF A>Amax THEN 111
110    GOTO 112
111    Amax=A
112    NEXT Theta
113    REM USE THE MAX ACCEL TO FIND THE MAX TORQUE AT EACH INCREMENT OF SPEED
115    T=M*Amax*R
116    REM************************************************
117    REM OUTPUT THE DATA TO A STORAGE FILE
120    OUTPUT @Stroke;Rpm," ",T
130    PRINT "TORQUE =";T," ","RPM=";Rpm
140    NEXT Rpm
150    ASSIGN @Stroke TO *
160    END
```

Appendix #4 *Vendor Information*

Thomson hardened-and-ground 60 Case steel shaft, grouting, and shaft supports

Table 37 — Metric diameter solid 1060 steel 60 Case shaft (Rockwell 60-65C)

Nominal Dia. (mm)	Class M Tolerances* (inch)	Max.** Length (feet)	Minimum Hardness Depth (inch)	Weight per Inch of Length (lb)	Price per Inch of Length ($)
5 mm	.1969/.1965	8	.040	.009	41
8 mm	.3150/.3146	14	.040	.022	46
12 mm	.4724/.4720	14	.060	.050	49
16 mm	.6299/.6295	14	.060	.088	57
20 mm	.7874/.7869	14	.060	.138	75
25 mm	.9843/.9838	14	.080	.216	97
30 mm	1.1811/1.1806	14	.080	.311	1.35
40 mm	1.5748/1.5743	14	.080	.553	1.67
50 mm	1.9685/1.9679	14	.100	.864	3.75
60 mm	2.3622/2.3615	14	.100	1.240	4.81
80 mm	3.1496/3.1489	14	.100	2.210	6.75

*Please specify Tolerance Class on order.
**For longer lengths, shafts may be joined at factory. Contact your Thomson distributor for a quotation.

Table 38 — BALL-GROOVE* 1060 steel 60 Case shaft (Rockwell 60-65C)

Nominal Dia. (inch)	Tolerances Class "G" (inch)	Maximum Length (inch)	Minimum Hardness Depth (inch)	Price per Inch of Length ($)
1/4	.2495/.2490	45	.040	.59
3/8	.3745/.3740	45	.040	.65
1/2	.4995/.4990	45	.060	.75
5/8	.6245/.6240	45	.060	.83
3/4	.7495/.7490	45	.060	.90
1	.9995/.9990	45	.080	1.20

*For use only with Thomson SUPER BALL BUSHING linear bearings.

Table 39 — GROUTING

Description	Quantity	Volume	Price ($)
Waystone grout	25-lb. pkg. 100-lb. drum	14 cu.in/lb.	24.50 56.50
Devcon F	1-lb. pkg.	15.5 cu.in/lb.	24.00

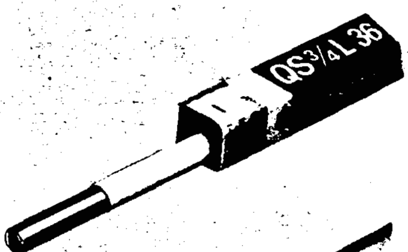


Table 40 — QUICK™ Shaft 1060 steel 60 Case Standard Length shafting (Rockwell 60-65C)

Part Number	Nominal Dia. (inch)	Class L Tolerance (inch)	Standard Length (inch)	Minimum Depth of Hardness (inch)	Weight per inch of length (lb.)	Unit Price ($)
QS 1/4 L 24	1/4	.2495/.2490	24	.040	.014	12.51
QS 3/8 L 24	3/8	.3745/.3740	24	.040	.031	13.50
QS 1/2 L 24	1/2	.4995/.4990	24	.060	.055	14.00
QS 5/8 L 24	5/8	.6245/.6240	24	.060	.086	15.74
QS 3/4 L 36	3/4	.7495/.7490	36	.060	.125	21.59
QS 1 L 36	1	.9995/.9990	36	.080	.222	27.56

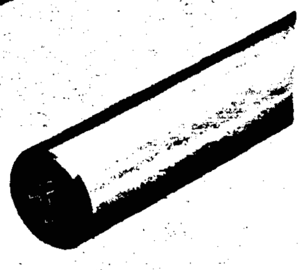


Table 41 — Class XL Shafts for Extra Rigid Series XR System

Nominal Diameter (inch)	Tolerance Class XL	Maximum Length (ft.)	Minimum Depth of Hardness (inch)	Weight per inch of length (lbs.)	Price per inch of length ($)
2	1.9994/1.9991	14	.100	.890	2.99
3	2.9992/2.9989	14	.100	2.003	5.44
4	3.9988/3.9983	14	.100	3.560	8.93

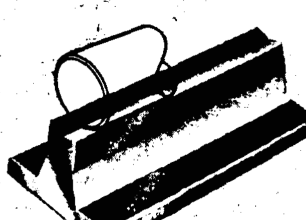


Table 42 — Shaft support rails (standard 24-inch lengths)

Part Number	Nom. Shaft Dia. (inch)	Unit Price ($) Based on Quantity Ordered					
		1-9	10-29	30-49	50-69	70-99	100+
SR-8	1/2	17.72	17.21	16.24	15.24	14.28	13.53
SR-10	5/8	20.10	19.45	18.22	17.01	16.03	15.08
SR-12	3/4	22.86	22.12	20.63	19.37	18.11	16.91
SR-16	1	27.30	26.43	24.66	22.94	21.27	19.73
SR-20	1-1/4	36.43	35.19	32.73	30.51	28.33	26.17
SR-24	1-1/2	43.19	41.81	38.98	36.27	33.55	31.13
SR-32	2	63.03	61.08	57.23	53.44	49.76	46.21

*For lengths other than standard contact factory.

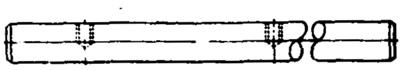


Table 43 — Class PD 1060 steel 60 Case shaft with drilled-and-tapped mounting holes (Rockwell 60-65C)

Nominal Dia. (inch)	Tolerances Class PD (inch)	Maximum Length (inch)	Hole Spacing* (inch)	Standard Thread Size	Price per Inch of Length ($)
1/2	.4995/.4990	166	4	6-32	1.56
5/8	.6245/.6240	178	4	8-32	1.61
3/4	.7495/.7490	178	6	10-32	1.66
1	.9995/.9990	178	6	1/4-20	1.85
1-1/4	1.2495/1.2490	178	6	5/16-18	2.18
1-1/2	1.4994/1.4989	178	8	3/8-16	2.46
2	1.9994/1.9987	178	8	1/2-13	3.93

Specify location of first hole when ordering. Set-up charge $20.00

Thomson SUPER BALL BUSHING linear bearing pillow blocks.

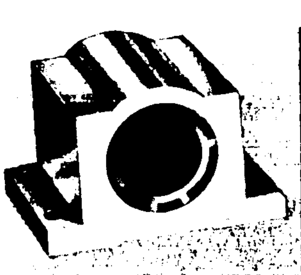


Table 5 — Fixed-diameter SUPER BALL BUSHING linear bearing pillow blocks

Part Number	Nom. Shaft Dia. (inch)	Unit Price ($) Based on Quantity Ordered					
		1-9	10-29	30-49	50-69	70-99	100+
SPB-4	1/4	25.12	24.40	22.96	21.56	20.16	18.81
SPB-6	3/8	26.26	25.50	24.01	22.53	21.07	19.64
SPB-8	1/2	28.23	27.42	25.81	24.23	22.66	21.14
SPB-10	5/8	32.16	31.15	29.12	27.13	25.16	23.38
SPB-12	3/4	34.15	33.01	30.75	28.54	26.60	24.62
SPB-16	1	48.76	47.21	44.06	41.04	38.19	35.38
SPB-20	1-1/4	73.40	70.99	66.29	61.62	57.07	52.75
SPB-24	1-1/2	92.38	89.40	83.53	77.78	72.09	66.51
SPB-32	2	140.09	135.62	126.83	118.17	109.60	101.33

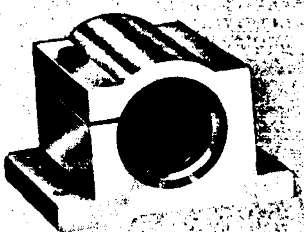


Table 6 — Adjustable-diameter SUPER BALL BUSHING linear bearing pillow blocks

Part Number	Nom. Shaft Dia. (inch)	Unit Price ($) Based on Quantity Ordered					
		1-9	10-29	30-49	50-69	70-99	100+
SPB-4-ADJ	1/4	28.29	27.46	25.82	24.22	22.64	21.17
SPB-6-ADJ	3/8	29.54	28.70	26.96	25.32	23.64	22.12
SPB-8-ADJ	1/2	31.78	30.86	29.02	27.21	25.44	23.79
SPB-10-ADJ	5/8	35.70	34.55	32.31	30.28	28.27	26.35
SPB-12-ADJ	3/4	37.67	36.52	34.24	32.02	29.81	27.65
SPB-16-ADJ	1	55.67	53.78	49.96	46.23	42.56	38.96
SPB-20-ADJ	1-1/4	83.06	80.29	74.92	69.62	64.38	59.25
SPB-24-ADJ	1-1/2	103.33	99.89	93.16	86.61	80.28	74.05
SPB-32-ADJ	2	158.30	153.19	143.05	133.13	123.51	114.06

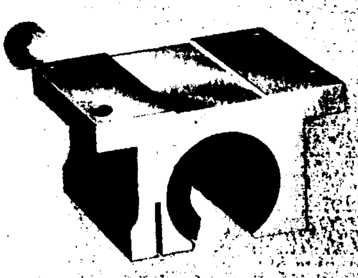


Table 7 — Open type SUPER BALL BUSHING linear bearing pillow blocks

Part Number	Nom. Shaft Dia. (inch)	Unit Price ($) Based on Quantity Ordered					
		1-9	10-29	30-49	50-69	70-99	100+
SPB-8-OPN	1/2	33.46	32.41	30.37	28.35	26.40	24.62
SPB-10-OPN	5/8	38.86	37.62	35.11	32.77	30.55	28.38
SPB-12-OPN	3/4	41.42	40.14	37.64	35.18	32.78	30.38
SPB-16-OPN	1	59.71	57.67	53.66	49.70	45.79	41.98
SPB-20-OPN	1-1/4	89.02	86.09	80.25	74.51	68.85	63.37
SPB-24-OPN	1-1/2	108.81	105.28	98.30	91.41	84.68	78.21
SPB-32-OPN	2	166.85	161.40	150.77	140.36	130.09	120.00

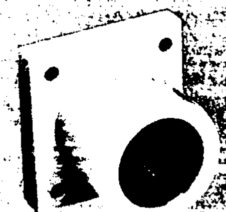


Table 8 — Fixed diameter SUPER BALL BUSHING flange blocks

Part Number	Nom. Shaft Dia. (inch)	Unit Price ($) Based on Quantity Ordered					
		1-9	10-29	30-49	50-69	70-99	100+
SFB-8	1/2	34.97	33.94	32.06	30.02	28.14	26.27
SFB-12	3/4	41.28	40.07	37.87	35.45	33.23	31.01
SFB-16	1	56.38	54.74	51.72	48.41	45.38	42.35

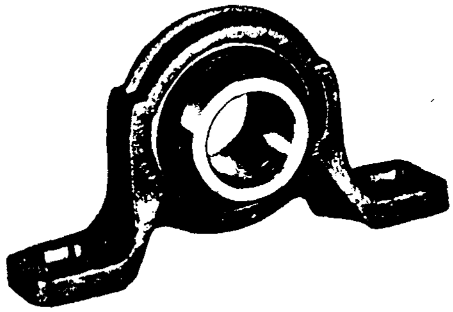
Table 8A — Open type Rigid SUPER BALL BUSHING pillow blocks

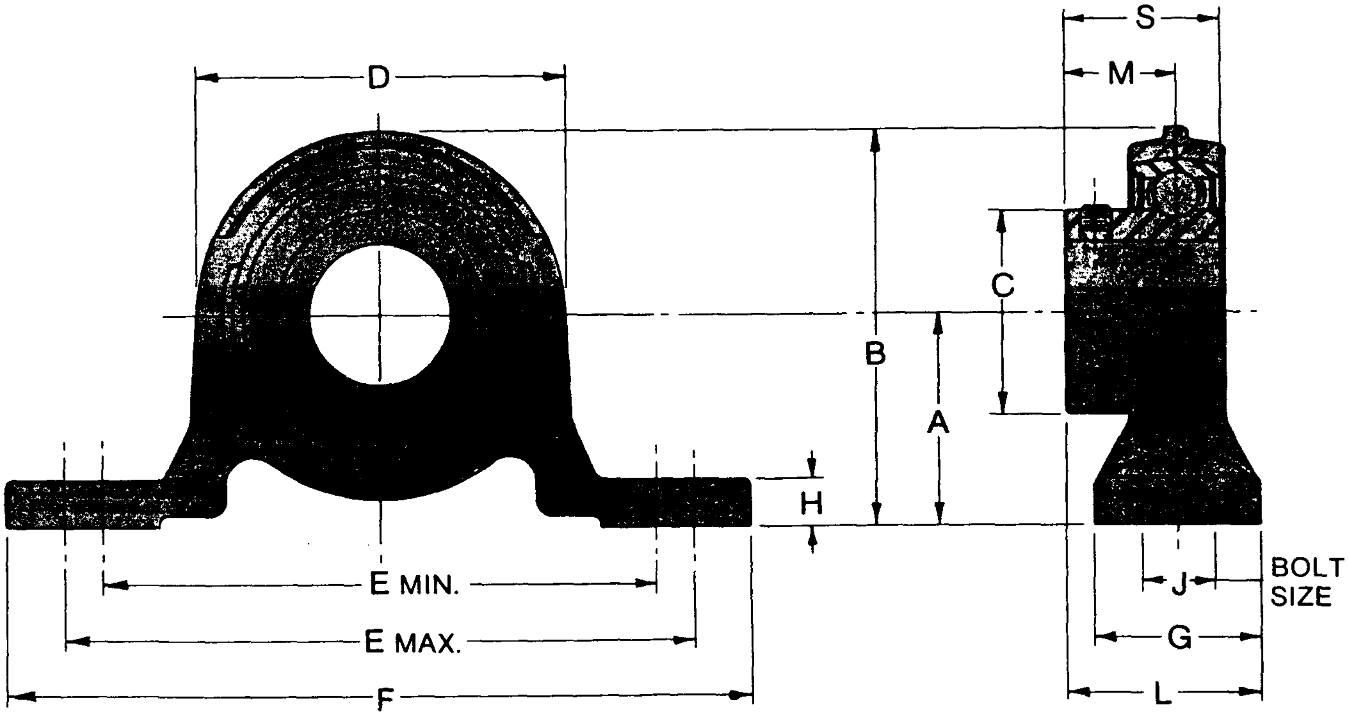
Part Number	Nom. Shaft Dia. (inch)	Unit Price ($) Based on Quantity Ordered					
		1-9	10-29	30-49	50-69	70-99	100+
RSPB-12-OPN	1/2	59.97	58.12	56.33	50.93	47.47	43.99
RSPB-16-OPN	3/4	74.44	71.90	69.38	61.95	57.08	52.32
RSPB-24-OPN	1	142.91	138.27	133.68	120.07	111.22	102.73

Material: Malleable iron

All Thomson products are sold exclusively through authorized distributors. For the name of your local distributor, call 1-800-645-9357.

- Non-relube type.
- Setscrew locking.
- Self-aligning replaceable bearing.
- Damage resistant contact seals.
- Precision bored Malleable Iron Housing.





PB100 SERIES — FOR HIGH SHAFT HEIGHTS

1/2, 5/8	1 3/16	2 5/32	.961	1 15/16	3 7/32	3 21/32	4 7/16	1	11/32	3/8	1 1/8	.630	.886	.6
3/4	1 5/16	2 7/16	1.165	2 1/4	3 17/32	3 31/32	4 3/4	1 1/16	3/8	3/8	1 15/64	.709	1.004	.8
7/8, 15/16, 1	1 7/16	2 11/16	1.335	2 1/2	3 23/32	4 5/32	5	1 1/8	13/32	3/8	1 21/64	.768	1.063	1.0
1 1/16, 1 1/8, 1 3/16, 1 1/4S	1 11/16	3 5/32	1.606	2 15/16	4 7/16	5	5 15/16	1 5/16	13/32	1/2	1 33/64	.866	1.220	1.4
1 1/4, 1 5/16, 1 3/8, 1 7/16	1 7/8	3 9/16	1.843	3 3/8	4 23/32	5 9/32	6 1/4	1 3/8	7/16	1/2	1 11/16	1.004	1.378	1.9

Part Numbers are specified by "PB100" and Bore Size; Example: PB100 × 1 1/8.
For Load Ratings, see page 411. For Comparison Charts, see page 404.
For replacement inserts use B100 Adaptor Bearings, see page 416.

PB100L SERIES — FOR LOW SHAFT HEIGHTS

1/2, 5/8	1 1/16	2 1/32	.961	1 15/16	3 7/32	3 21/32	4 7/16	1	9/32	3/8	1 1/8	.630	.886	.6
3/4	1 1/4	2 3/8	1.165	2 1/4	3 17/32	3 31/32	4 3/4	1 1/16	5/16	3/8	1 15/64	.709	1.004	.8
7/8, 15/16, 1	1 5/16	2 9/16	1.335	2 1/2	3 23/32	4 5/32	5	1 1/8	11/32	3/8	1 21/64	.768	1.063	.9
1 1/16, 1 1/8, 1 3/16, 1 1/4S	1 9/16	3 1/32	1.606	2 15/16	4 7/16	5	5 15/16	1 5/16	3/8	1/2	1 33/64	.866	1.220	1.3
1 1/4, 1 5/16, 1 3/8, 1 7/16	1 13/16	3 1/2	1.843	3 3/8	4 23/32	5 9/32	6 1/4	1 3/8	13/32	1/2	1 11/16	1.004	1.378	1.9

Part Numbers are specified by "PB100L" and Bore Size; Example: PB100L × 7/8.
For Load Ratings, see page 411.
For replacement inserts use B100 Adaptor Bearings, see page 416.

[redacted]

SERIES: PB100, PB100L, PB150, PB150L PILLOW BLOCKS FB100, FB110, FB150, FB160 FLANGE BLOCKS B100 ADAPTOR BEARINGS

[redacted]

½, ⅝	1,150	675	535	465	425	395	370	350
¾	1,540	900	715	625	565	525	495	470
⅞, 15/16, 1	1,685	985	780	685	620	575	540	515
1⅛, 1 3/16, 1¼S	2,325	1,360	1,080	940	855	795	750	710
1¼, 1 5/16, 1⅜, 1 7/16	3,085	1,805	1,430	1,250	1,135	1,055	995	945

SERIES: 52SFB100, 62SFB100 FLANGETTES

[redacted]

⅞, 15/16, 1	800	800	780	685	620	575	540	515
1⅛, 1 3/16, 1¼S	1,100	1,100	1,080	940	855	795	750	710

Maximum Recommended Speed — 3500 RPM.
Recommended Operating Temperature Range — -25° to +225°F.
For ultimate life, snug fit or light press on shaft should be used.
For applications not covered in the table above, consult factory engineering.
For applications with shock loads or for other life requirements, refer to table below.
For applications involving severe shock loads, consult factory engineering.

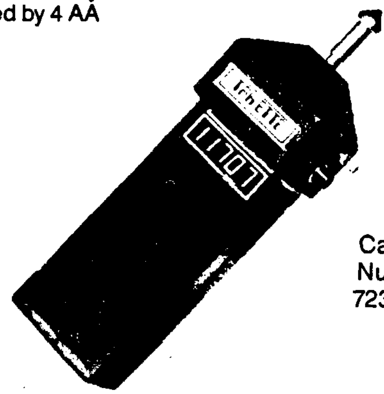
[redacted]

Steady Load	1.00	.941	.855	.794	.747	.709	.679	.630	.550	.500	.464	.437	.397	.368
Light Shock	.90	.847	.770	.715	.672	.639	.611	.567	.495	.450	.418	.393	.357	.331
Moderate Shock	.70	.659	.599	.556	.523	.500	.475	.441	.385	.350	.325	.306	.278	.258

Digital Contact Tachometer

- **Five Digit Readout to 20,000 RPM**
- **Accuracy: ±1 RPM**
- **Two Modes—RPM and Linear Speed in Ft/Min**
- **Battery Powered**

This hand held, contact wheel tachometer measures rotation to 20,000 RPM, or linear speed to 2,000 Ft/Min. It will give accurate readings from motors, gears, pulleys, fans, shafts, webs, conveyors, belts, paper rolls—almost anything that either rotates or moves linearly. Pushbutton memory holds last reading indefinitely and can recall it even after the tachometer has been shut off. To switch from RPM to linear speed measurement, just remove the rubber contact tip and replace it with the foot measuring wheel. Digital readout uses high visibility ⅜" LED's. The time base is 1 second, quartz crystal controlled. Accuracy is ±1 RPM. With dimensions of 8" x 2¼" x 1¼" HWD—and weighing only 12 ounces—the 7230FST is operated easily with just one hand. Power is supplied by 4 AA (1.5V) cells.



Catalog Number 7230FST

Each Unit Includes:

1. Foam-padded Carrying Case
2. Two Rubber Tips
3. Foot Measuring Wheel
4. Four Alkaline AA Cells

Ordering Information:

Catalog No.	Description	Price
7230FST-Y	Digital Contact Tachometer	$180.00

Digital Photo Tachometer

- **Requires No Contact with Moving Object**
- **Direct Reading to 30,000 RPM**
- **Accuracy: ±1 Digit Up to .03%**
- **Comes with Marking Tape and Discs**

This portable, no-contact tachometer measures RPM's at distances of ¼" to 30" from rotating objects, using a beam of light and reflecting tape. It is a safe way to measure moving machinery at a distance and is also useful in hard-to-get-at spots. The light source on 24" coiled cord is particularly handy in those difficult locations. A "target eye" in the readout lights up when contact has been made and the instrument is ready for measurement. Range is 10 to 30,000 RPM, without the need for range switching. RPM's are read directly from the 5-digit, ⅜" high LED display. A built-in memory holds the last value indefinitely. This digital photo tachometer can be used to measure RPM's in centrifuges, motors, drills, lathes, or any rotating object within the range of the instrument. The 7240FST has a pushbutton ON/OFF switch and is powered by one 1.5V D cell and four 1.5V "AA" batteries. Overall dimensions are 8½" x 4½" 2" HWD; the unit weighs 1.5 pounds.

Catalog Number 7240FST

Each Unit Includes:

1. Foam-padded Carrying Case
2. 1" Wide Reflection Tape, 2 Feet Long
3. Ten Discs
4. One 1.5V Size D Battery
5. Four 1.5V "AA" Batteries

Ordering Information:

Catalog No.	Description	Price
7240FST-Y	Digital Photo Tachometer	$295.00

Digital Stroboscope/Tachometer

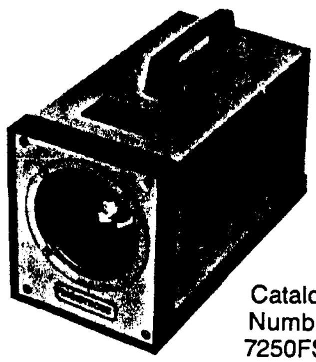
- **Freezes Motion and Provides Instant RPM**
- **Digital Readout with ⅜" LED's**
- **Flash Rate: 200 to 12,000 Flashes per Minute. Tach: 200 to 12,000 RPM**
- **Accuracy: ±1 Digit**

This unusual combination of a high-intensity strobe and easy-to-use digital tachometer is well suited to a variety of tasks. Use it to check and analyze motion and to observe objects like bottles, cans, and packages on conveyor belts; to check out belt slippage; to verify the register of printing presses; and to determine the speed of motors, gears, pulleys, fans, and shafts. A simple turn of the control knob adjusts the light flashes into synchronism with the observed RPM of the object.

Flash rate for the strobe is 200 to 12,000 flashes per minute. Flash color is Xenon white, 6500°K. The duration is 10 to 25 microseconds, the most efficient rate for the human eye. The lens is plastic and there is a mirror-type reflector.

Basic range of the tachometer is 200 to 12,000 RPM. However, by using harmonics, speeds from 12,000 to 100,000 RPM can be easily measured.

This multi-purpose instrument is housed in a steel case that weighs 7.75 pounds and measures 11" x 6½" x 5½". It utilizes a 115 volt, 50/60 Hz power source.



Catalog Number 7250FST

Ordering Information:

Catalog No.	Description	Price
7250FST-Y	Digital Stroboscope/Tachometer	$467.00

Appendix #5 *Miscellaneous*

Lunar Soil Discussion

- The lunar surface has many volcanic features and volcanic rocks.
- The iron-rich mare soils melt at low temperatures (≈1,200°C).
- The highland soils melt at ≈1,400°C.
- Mare soils have a low melt viscosity of ≈ 10-100 poises.
- Deep drilling and coring will be very time-intensive.
- Lubricating fluids for the core tubes and bits is essential.
- Example: 100m of core tubing, 10 cm in diameter and full of rock and regolith would possess a mass of roughly 2 metric tons.
- Strengths of materials are less of a problem for lifting and supporting, although not for withstanding changes in momentum.
- The lunar surface abounds in fine dust.
- The lunar soil is described as a fine rock flour produced by past impacts of meteorites.
- Some of the particles are glassy agglutinates.
- This rock flower contains fragments ranging in size from dust to boulders.
- Soil from mare sites are made mainly from crushed lunar basalt.
- In uncontaminated hard vacuum the surface friction and strengths of the rocks are higher than our terrestrial rocks.
- The greater adhesive forces from higher surface energies and lack of water vapor will cause clogging during drilling.
- The increase in friction ranges from 1.5 to 60 times.
- It is suggested that compacted particulate minerals will require greater energy input to induce flow.
- Dust will affect visibility, operator ability, and equipment life.
- Airborne dust will have a linger dwell time because of the lunar conditions.
- Thermal expansion of the machine tool is possible because of the large extremes in temperature.

STEPPED AUGER JIG AND CONTROL DESIGN

Progress Report 1

Date: Thursday October 1, 1987.

The group had an informal meeting on Monday September 28. At this meeting we discussed the problem definition, what material should be initially researched, what kind of graphics could be generated, and formed a tentative schedule of completion for the project. We then divided the group into five sub-groups which worked on writing the problem definition, researching lunar soil and environment, researching designs of impact drills, generation of graphics, and the writing of the progress report. The group then began to brain storm on what format the controls should take. After some discussion we decided to drop the hydraulic format from last quarter and to go with mechanical and electronical control format. But we all agreed to be open to any other viable alternative.

The group had a second informal meeting on Thursday October 1 before the group's formal meeting. At this meeting we gathered all of the reports, research material, and graphics. We then discussed further on an initial design.

Jim Cika

Anthony Dean

Robert Dowd

Greg Horne

Ben Owens

Micheal Richardson

John Rockholt

Richard Verch

STEPPED AUGER JIG AND CONTROL DESIGN

Progress Report 2

Date: Thursday October 8, 1987.

The group had an informal meeting on Sunday, October 3. At this meeting we brainstormed and came up with several completely different ideas for our design. The first involves an electro-pnuematic device that is used in many commercial impact drills. The second idea uses the drill press concept adapted to provide a variable stroke and drill speed which can be computer controlled. The third idea is to use hydraulics to provide the impact motion, however we have only speculated on any ways to adapt this. Our next step is to come up with the best idea or combination of ideas and begin to finalize the design. We have also investigated the recipe for lunar soil. Finally, we are in the process of altering the auger design specs such as rotational speed and stroke length to accomodate gravitational effects on earth.

A simple graphic illustrating the drill press adaption is included with this weeks progress report. In addition, we have presented published literature that includes illustrations and specifications for the electro-pnuematic impact drill.

Jim Cika

Anthony Dean

Robert Dowd

Greg Horne

Ben Owens

Micheal Richardson

John Rockholt

Richard Verch

Progress Report

Week 4

This week we outlined a rough schedule for the time remaining this quarter. Continuied gaining background information on different systems with emphasis on the hydraulic design. A database search was done to obtain new, more detailed information on the components in our designs.

1. Cika, Jim - Read report done by last quarters design group. Studied their force analysis calculations and began analyzing the power requirements needed for our system.

2. Dean, Tony - Performed database search for references on hydraulic power, microprocces sor control, and mechanical design. Reviewed force analysis from the previous quarter.

3. Dowd, Robert - Studied versacad and helped with the hydraulic design drawing, and researched control logic of hydraulic systems.

4. Horne, Greg - Analyzed power requirements for the system which would drive the bit in the specified manner.

5. Owens, Ben - Researched the VSMF system for information on hydraulic control. Came up with a drawing for the hydraulic drill design on VersaCad.

6. Richardson,Micheal - Researched industrial construction tool manufactures. Obtained catalog of pneumatic powered tools with detailed specifications.

7. Rockholt, John - Performed general database search on hydralic system/controls. Performed spefic database search on Hydralics and Pneumatics Journal. Search vender VSMF system for companies that manufacture hydralic components. Studied hydralic system controls.

8. Verch, Richard - Continued design of prototype auger control. Looking at basic design which uses hydraulics to control both the reciprocating and rotory motion of the auger.

Progress Report

Week 5

This week we meet with several outside references including professors here at Tech. We then studied some of the suggested ideas and references to try to come up with a slider-crank design for the drill. The power requirements for the most extreme frequency to be tested were calculated.

1. Cika, Jim - Talked to Dr. Lipkin about a possible linkage design for obtaining the required longitudinal motion of our auger. He suggested a two bar design that would give the best mechanical advantage for the given constraints of our system. Looked at possible ways of driving this two bar linkage with rotary input.

2. Dean, Tony - Researched for references in library. Obtained two references on hydraulic power, Shigley's handbook on mechanical design, and a reference on microproccessor contol.

3. Dowd, Robert - Met with Dr. Ferri to get ideas for a possible linkage design. Researched transducers which would measure linear displacement as well as linear and angular velocity. Some good possibilities were discovered.

4. Horne, Greg - Continued analyzing the power requirements for the system which would drive the auger in the specified manner.

5. Owens, Ben - Met with Dr. Ferri to discuss the posibilities for a slider-crank type design & then researched some of the suggested references.

6. Richardson,Michael - Talked and began researching the use of slider-crank mechanism Set up meeting to talk to Dr. Ferry about the mechanism.

7. Rockholt, John - Continued work on hydralic system controls, and studied the application of a slider-crank mechanism.

8. Verch, Richard - Robert and I met with Dr. Ferri to investigate possible ways of obtain- ing the motions desired in the auger. A cam was suggested for the trans- lational motion. Looking for ideas and modifying existing ideas.

Stepped Auger Control Design

Progress Report

Week 6

This week we began preparing for the oral presentation. Each person researched different topics. No new revelations theis week.

1. Cika, Jim - Researched drillpress designs to see if any could be applied to our project. Not much was found in the area of design so this direction was abandoned. Also looked at reference materials on bearings.
2. Dean, Tony - Researched for various linkage designs. Began analysis on slider crank and scotch yoke linkages. Research iterative programs to analyze postion, velocity, and accelerations of linkages.
3. Dowd, Robert - Worked on getting plots of the relationships between some of our variables such as volume flow rate vs. frequency. Found that some changes need to be made in last quarters program to be useful to us. Also made some transparancies for today's oral presentation.
4. Horne, Greg - Wrote a basic program which calculated the required power for the specified frequency range. This program also generated data files of the power and corresponding frequency for various stroke lengths. From these data files a family of curves was plotted for the Power Analysis graph (see attached graph).
5. Owens, Ben - Due to a week of mid-terms, I didn't do much other than prepare what I was going to say in the oral report.
6. Richardson,Michael - Started researching the hydraulic cylinder and ball spline design. Talked with Doctor Johnson about other linkage designs.
7. Rockholt, John - Continued work on the application of a slider-crank mechanism with extra cosideration placed on the scotch-yoke type design.
8. Verch, Richard - Did calculations on possible cam profiles. Decided that if square wave input was desired, that the cam is not the best way of accomplishing this Prepared a graphical cam analysis. Worked on section of oral report.

Stepped Auger Control Design

Progress Report

Week 7

This week we have decided to go with the yoke design because of its relative ease to build and control and its mechanical simplicity. We came up with drawings, power requirements, and a force analysis of the system. We also studied the motor requirements for the parameters involved.

1. Cika, Jim - Researched bearing design and made a rough sketch of the coupling joint for our drill rig.
2. Dean, Tony - Continuing analysis on slider crank and scotch yoke linkages. Trying to write computer programs analyzing postion, velocity, and acceleration of linkages. Made rough drafts of preliminary designs. Began analyzing forces in scotch yoke linkage.
3. Dowd, Robert - Calculated the approximate motor size for several different parameters for our test apparatus. Also, concentrated on the type of structure needed to support the apparatus.
4. Horne, Greg - Investigated ways of counterbalancing the oscillitory motion of the jig.
5. Owens, Ben - Researched the motor requirements of the system and after consulting people in the Electrical Engineering Department, I found that a series DC motor would be best for the torque and rpm requirements our system needs.
6. Richardson,Michael - Started researching the hydraulic cylinder and ball spline design. Talked with Doctor Johnson about other linkage designs.
7. Rockholt, John - Continued work on the application of a slider-crank mechanism with extra cosideration placed on the scotch-yoke type design.
8. Verch, Richard - Did calculations on possible cam profiles. Decided that if square wave input was desired, that the cam is not the best way of accomplishing this Prepared a graphical cam analysis. Worked on section of oral report.

Stepped Auger Control Design

Progress Report

Week 8

This week we searched through the VSMF files to find manufacturer's products that we will possibly use in our design. From the manufacturer's catalogs we obtained the necessary product descriptions and specifications. Also, we continued on the power requirements and linkage analysis of the system. Finally, we continued to develop our design concept.

1. Cika, Jim - I searched the Magi, Trend, and Gtec databases for information on bearings and support beams. Some books were looked up but as of now no pertinent information has been found.

2. Dean, Tony - I searched through VSMF files for manufacturers specs on pillow cushions, guide bearings, and bearing rings. Derived equations for power without balancing and power with balancing. Began modified drafts of the scotch yoke design.

3. Dowd, Robert - I conducted a VSMF search for products we will need and their specifications. Also, I wrote a program to plot the power required by the mechanical system.

4. Horne, Greg - Searched vendor cataloges for shaft supports and bearings. The appropriate pillow blocks were found in the McGill Mfg. catalog for supporting the wheel drive shaft. However, I was unable to locate guides for drill shaft, which not only require rotational movement but also translational motion.

5. Owens, Ben - Searched the VSMF system for information and specifications of roller bearings needed between the drill shaft and the wheel. I also continued research on the motor requirements needed.

6. Richardson,Michael - Started researching the hydraulic cylinder and ball spline design. Talked with Doctor Johnson about other linkage designs.

7. Rockholt, John - Continued work on the application of a slider-crank mechanism with extra cosideration placed on the scotch-yoke type design.

8. Verch, Richard - Found a manufacturer of plastics that produces clear plastic tubing which could be used for the auger sleeve, thus allowing us to observe the action of the particles. The company is located here on North Ave.

Stepped Auger Control Design

Progress Report

Week 9

This week we began making drawings of the different components of our design. Also, we began a rough draft of the final paper. Finally, we continued to work on the analysis of the design structure.

1. Cika, Jim - Learned to use Geodraw and worked on a drawing of our our experimental test jig.
2. Dean, Tony - Began to learn GEODRAW and to use it to draft the final design. I also re-worked the power analysis giving a more complete estimation of power needed.
3. Dowd, Robert - I began writing the rough draft of the paper including the problem statement and abstract. I also continued work on the type of motor to be used in the design.
4. Horne, Greg - Began formulating rough draft of final report and learned how to use Geodraw.
5. Owens, Ben - Did CAD drawings on the roller bearings through researching the spec- ifications on the VSMF file system. I also continued research on the motor requirements needed.
6. Richardson,Michael - Began researching the VSMF file system for information on sphere rollers. Specifications were found so drawings can be made.
7. Rockholt, John - Met with Elain Wagner in the library to discuss the topic search. Also began to learn Geodraw.
8. Verch, Richard - Found a manufacturer of plastics that produces clear plastic tubing which could be used for the auger sleeve, thus allowing us to observe the action of the particles. The company is located here on North Ave.